THE DESIGN, FEASIBILITY, AND OPTIMIZATION OF AN AMMONIA BOTTOMING CYCLE FOR POWER GENERATION

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ABSTRACT

The economic feasibility of utilizing an ammonia bottoming cycle to improve the efficiency of a power plant is studied. The operating parameters of the bottoming cycle are examined to determine the optimum design binary cycle from a lifecycle costs viewpoint.

The performance of the combined cycle is compared with that of a representative 'modern' steam plant: TVA's Bull Run. The analysis is based on current economic factors and the climatic conditions at the Bull Run site in Clinton, Tennessee. The optimum combined cycle is $98 million more profitable than the 'Bull Run' plant over the expected lifetime of the plant. The combined cycle also consumes 2.7% less fuel than the unmodified steam plant.

Optimization results are also presented for a range of environmental conditions at the plant location.

The use of an ammonia bottoming cycle is economically as well as technically feasible provided the ambient water temperature is below 69°F.

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A	Area
A_B	Boiling region heat transfer area
A_{FLOW}	Area perpendicular to flow
A_{NB}	Non-boiling region heat transfer for area
A_T	Total heat transfer surface area
C	Specific heat at constant pressure
C_p	Specific heat at constant pressure
D	Tube diameter
F	Chen factor
f	Friction factor
f_{TP}	Two-phase friction factor
g	Acceleration of gravity
g_o	32.2 ft-lbm/lbf-sec^2
h	Enthalpy
h	Heat transfer coefficient
h'_{fg}	Modified enthalpy of vaporization
HHV	Higher Heating Value
h_p	Horsepower
HP	High-Pressure
i	Interest rate
IP	Intermediate-Pressure
k	Thermal conductivity
k_f	Thermal conductivity
kw	Kilowatts
L	Tube length
L_B	Boiling region tube length
LF	Load factor
LMTD	Log mean temperature difference

L_{NB}	Non-boiling region tube length
LP	Low Pressure
$\dot{m}$	Mass flow rate
MW	Megawatts
n	Number of tubes in vertical row
n	Number of years of plant life
NH_3	Ammonia
P	Pressure
PL	Power loss due to turbine backpressure
P_{NH_3}	Ammonia cycle power
PP	Condenser Pumping power
psi	Pounds per square inch
P_{ST}	Steam cycle power
P_T	Turbine power
Q	Heat transfer
Q_A	Heat added
Q_B	Heat transfer in boiling region
Q_{NB}	Heat transfer in non-boiling region
Q_R	Heat rejected
Re	Reynolds number
Re_{sp}	Single-phase Reynolds number
Re_{TP}	Two-phase Reynolds number
S	Nucleate boiling suppression factor
s	Entropy
SC/AB	Steam condenser/ammonia boiler
T	Temperature
TAV	Entropy averaged temperature of heat addition to ammonia
T_c	Condensing temperature
TF(1)	Ammonia boiling temperature
TF(2)	Ammonia boiler inlet temperature
TF(5)	Ammonia condensing temperature

T_H	Average temperature of heat addition
T_L	Average temperature of heat rejection
T_{NH_3}	Ammonia temperature
T_S	Steam condensing temperature
T-S	Temperate-entropy
T_{wall}	Tube wall temperature
U	Overall heat transfer coefficient
V	Velocity
v	Specific volume
W_P	Pump work
x	Thermodynamic quality
$\bar{x}$	Average quality in turbine stage
x_e	Turbine exit quality
X_{tt}	Martinelli-Nelson factor

SUBSCRIPTS

BR	Bull Run
cc	Combined cyle
e	Extraction
f	Property at saturated liquid state
fg	Property change associated with vaporization
g	Property at saturated vapor state
v	Property at saturated vapor state

GREEK SYMBOLS

Δh	Enthalpy difference
ΔP	Pressure difference
ΔT	Temperature difference
$\Delta\bar{T}$	Difference between T_S AND TAV
ΔT_{LM}	Log mean temperature difference
ΔT_W	Temperature difference between condensing fluid and tube wall section
η_a	Auxiliary efficiency
η_b	Boiler efficiency

η_c	Cycle efficiency
η_g	Generator efficiency
η_{is}	Turbine isentropic efficiency
η_M	Mechanical efficiency
η_{OA}	Overall cycle efficiency
η_p	Pump efficiency
η_{ST}	Steam cycle thermal efficiency
η_{steam}	Steam cycle thermal efficiency
η_{th}	Thermal efficiency
μ	Dynamic Viscosity
$\bar{\mu}$	Average dynamic viscosity
ρ	Density
σ	Surface tension

I. INTRODUCTION

During the past few years, as the demand for electrical power has increased, mankind has become increasingly concerned over several important factors which will have impact on how electrical power will be generated in the future. The first of these considerations is the dramatic increase in the price of fuel and the realization that our fuel resources are limited. Secondly, we have become more concerned over how the electrical power generating plant affects our environment. The specific environmental considerations are how the fuel combustion process pollutes the air and how the heat which the plant rejects affects the temperature of the heat sink, usually a lake or river.

In order to partially overcome these problems a plant with a higher thermal efficiency than presently exists is desired. The higher efficiency plant would, for the same power output, burn less fuel and therefore exhaust fewer combustion products to the atmosphere. Furthermore, as the plant efficiency increases the amount of heat rejected per unit power output decreases.

The problem is: How to increase the efficiency of generating electric power?

There are many ways to design a highly efficient power plant. Some of the proposed 'advanced' cycles are both expensive and exotic. The possibilities include:

(1) Fuel-cells

(2) MHD

(3) Combined steam/gas turbine cycles

(4) Liquid metal topping cycles.

(5) Bottoming cycles utilizing a refrigerant-type working fluid.

In this thesis, the bottoming cycle will be examined.

The thermal efficiency of an ideal cycle $\eta_{th} = 1 - \frac{T_L}{T_H}$ (Equation 1)

where T_L is the average temperature at which heat is rejected and T_H is the average temperature of heat addition to the cycle.

The purpose of a bottoming cycle is to increase the thermal efficiency of a cycle by lowering T_L. Conventional steam plants are unable to take full advantage of a low heat rejection or condensing temperature due to:

(1) The large specific volume of steam at low condensing temperatures and pressures.

(2) The increased moisture content in the form of droplets as the condensing temperature drops.

The above considerations require large, costly, and relatively inefficient low pressure turbine stages.

In order to overcome these limitations and to take full advantage of a low temperature heat sink, another Rankine cycle using a different working fluid could be added to a steam plant. This 'bottoming' cycle would receive the energy which the steam plant rejects and would have the capability of lowering condensing temperatures.

The selection of a working fluid for the 'bottoming' or sub-position cycle cannot be accomplished by a closed form equation, but rather by a series of trade-offs. The desired properties of the working fluid are:

(1) Have a relatively low specific volume at the condensing temperatures to be considered.

(2) Be a liquid at ambient temperatures, so it can be pumped.

(3) Be non-toxic and non-corrosive.

(4) Have a low specific heat and a large latent heat of vaporization over the temperature range to be considered. (This results in the average temperature of heat addition being as high as possible).

(5) Have a condensing pressure above atmospheric pressure.

(6) Have a high speed of sound at turbine exit conditions.

(7) Have good heat transfer properties.

(8) Be chemically stable and non-flammable.

(9) Be inexpensive and plentiful.[1]

No known substance satisfied all of these requirements, but Ammonia (NH_3) seems to be the best overall choice.

The feasibility of the ammonia bottoming cycle will be evaluated by investigating a representative 'modern' steam plant modified to include the sub-position cycle. The thermal efficiency, capital, and operating costs of the binary cycle plant will be compared with those of the unmodified 'modern' plant. With these factors in mind an optimum design combined cycle plant will be generated, which will result in maximum savings to the owner/operator over the expected lifetime of the plant.

II. LITERATURE SURVEY

The concept of a bottoming cycle is not new. In 1961, Aronson[2] proposed such a binary-vapor cycle, utilizing Freon -12 as the working fluid. His analysis showed a heat rate improvement of 1-2% over that of a 'modern' steam plant as well as establishing the economic feasibility of the cycle.

In 1969, Wood[3] stated that a binary-vapor bottoming cycle was not feasible, citing available energy loss due the temperature difference across the steam condenser/secondary fluid boiler.

The Aronson proposal has been supported by Steigelman et al[4], who demonstrated the desirability of a binary-cycle using cooling towers provided the ambient air temperature was low enough. Furthermore, El Ramly and Budenholzer[5] professed the advantages of a bottoming cycle to operate in conjunction with a nuclear power plant.

Perhaps the most comprehensive study to date has been that of Slusarek[6], in which the feasibility of the ammonia bottoming cycle was studied in detail. The Slusarek report included a thorough component design, an economic analysis, and an engineering optimization. While the majority of the report is accurate, two significant shortcomings must be noted:

(1) The heat transfer coefficients in the ammonia boiler were calculated using inappropriate correlations.

(2) The impact of the steam condensing temperature and the temperature difference between the main and bottoming cycles were not adequately examined.

In this study, the operating parameters of a bottoming cycle will be studied and optimized in order to prepare a preliminary design of the best plant from an engineering as well as economic viewpoint.

III. THE COMBINED CYCLE

As outlined in chapter one, the bottoming cycle utilizes the heat rejected by a steam plant as its energy source. A simplified diagram of a combined steam-ammonia plant, hereafter called the 'combined' cycle, is shown in figure one.

To carry out the analysis of the feasibility and economic optimization of the combined cycle, an existing 'modern' steam plant was modified to include a bottoming cycle. The Tennessee Valley Authority's 'Bull Run' plant was selected for this role.

Bull Run is a 900 MW, coal fired, supercritical steam plant which first went into operation in the mid 1960's. The plant features a boiler exit temperature of 1000°F and a single reheat to the same temperature. The design condenser pressure is 1.5" of mercury, which corresponds to 91.7°F and .74 psi. The thermal efficiency of Bull Run, excluding the boiler losses is 45.78%. Including the boiler efficiency of 89%, the cycle efficiency is 40.74%.[7]

The Bull Run plant flow diagram and the Temperature vs. Entropy diagram are shown as figures two and three respectively.

The ammonia bottoming cycle is a simple Rankine cycle. Feedheating must be utilized since the feasibility analysis is sensitive to small changes in the efficiency of the bottoming cycle.[8]

Based on earlier studies, a four stage turbine for the ammonia cycle seems to be a logical choice. Consequently, the cycle will have three stages of feedheating. Data from steam plants indicates that the maximum efficiency

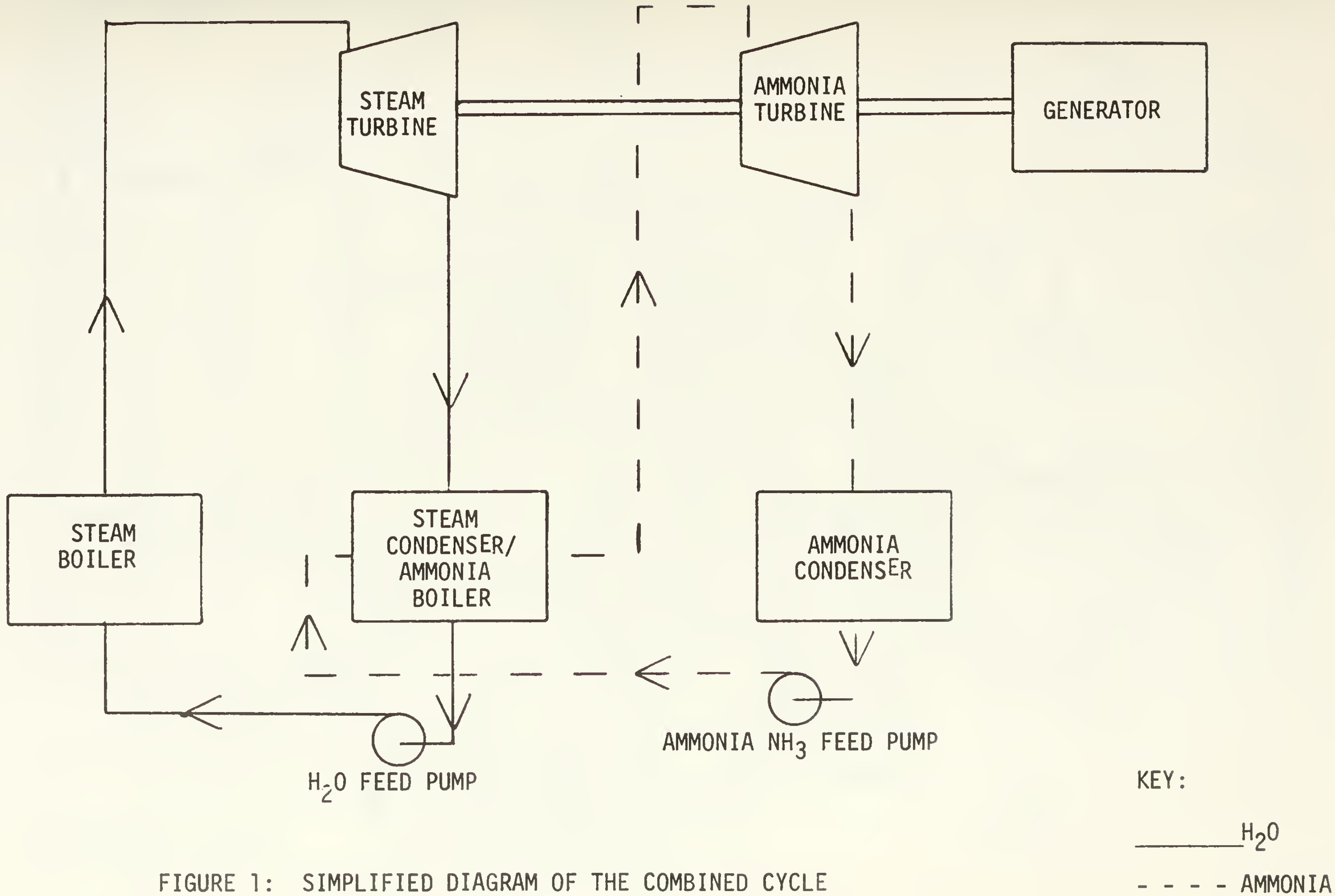


FIGURE 1: SIMPLIFIED DIAGRAM OF THE COMBINED CYCLE

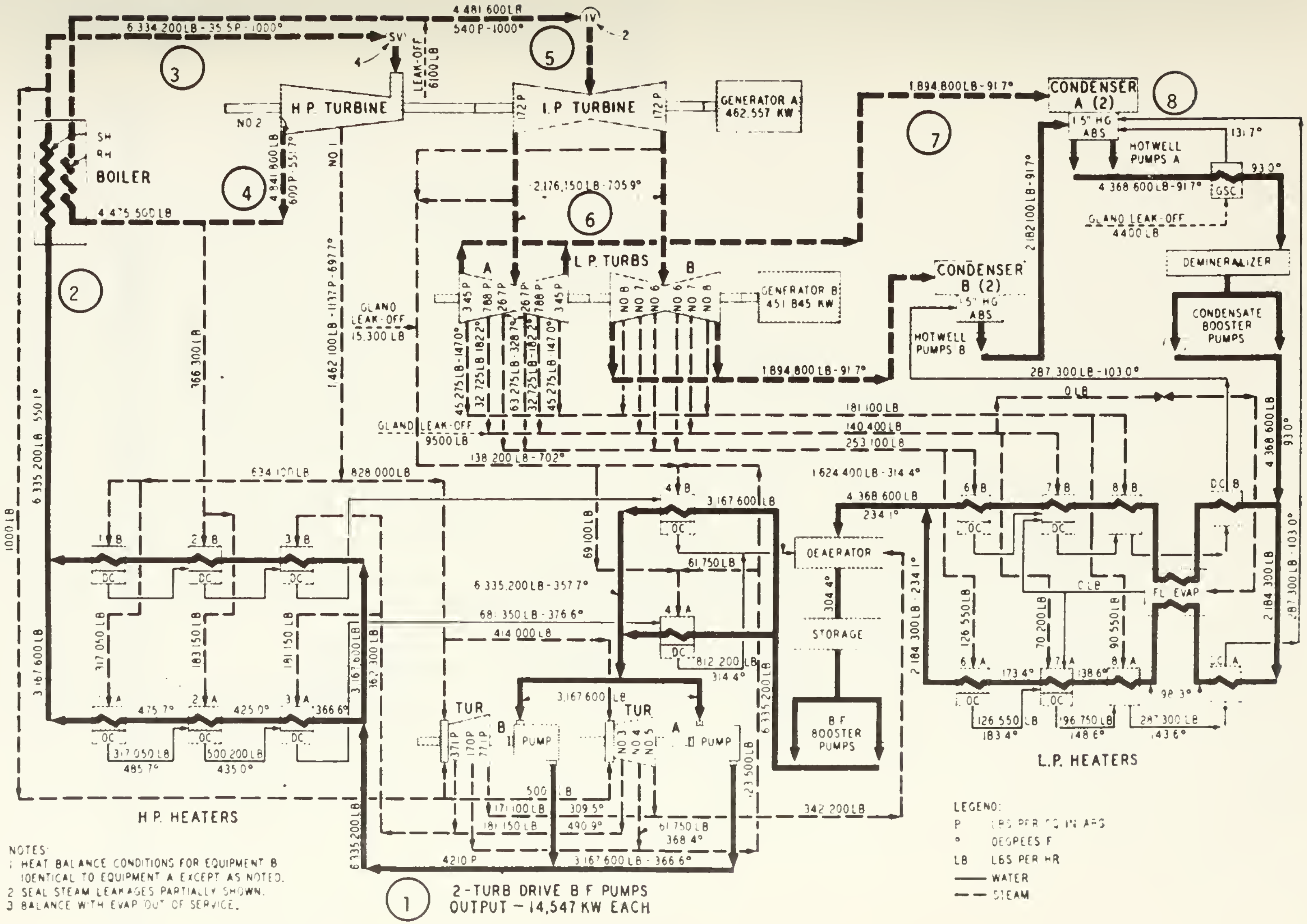


FIGURE 2. HEAT BALANCE DIAGRAM OF TVA'S BULL RUN (FROM REFERENCE 14)

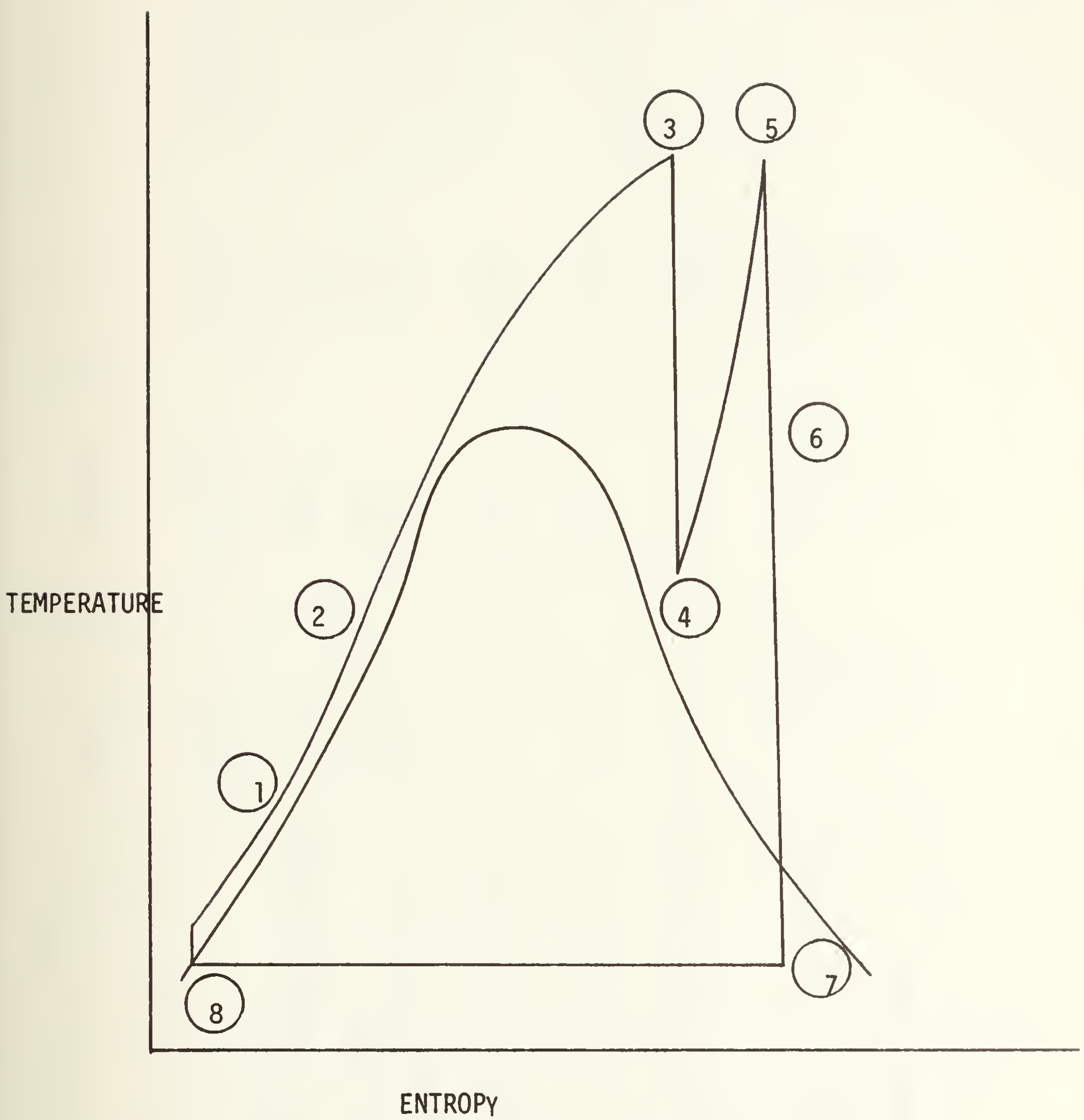


FIGURE 3 TEMPERATURE - ENTROPY DIAGRAM OF TVA'S BULL RUN PLANT

TABLE 1: TENNESSEE VALLEY AUTHORITY'S BULL RUN

STEAM NUMBERS REFER TO FIGURES 2 and 3

STREAM	NO.	DESCRIPTION	PRESSURE (psi)	TEMPERATURE (°F)	ENTHALPY (BTU/lbm)	MASS FLOW RATE (lbm/hr x 10^6)
Water	1*	Boiler feed pump discharge	4210	366.6	345.6	6.33
Water	2*	Boiler inlet	4210	550.1	544.9	6.34
Steam	3	Boiler exit/HP turbine inlet	3515	1000	1420.8	6.34
Steam	4	HP turbine exit/reheater inlet	600	551.7	1255.4	4.84
Steam	5	Reheater exit/IP turbine inlet	540	1000	1519.6	4.48
Steam	6	IP turbine exit/LP turbine inlet	172	705.9	1381.4	4.48
Water and Steam	7	LP turbine exit/condenser inlet	0.74	91.7	1002.6	3.79
Water	8	Condensed liquid	0.74	91.7	59.8	3.79

* INCLUDES FEEDHEATING

increase for three stage feedheating occurs when the working fluid temperature is elevated three-quarters of the difference between the condensing and evaporating temperature by feedheating.[9]

The Temperature-Entropy diagram of the bottoming cycle is shown as figure 4. Note that the ammonia is not superheated. In order to maximize the efficiency of the combined cycle, the average temperature of heat addition to the ammonia cycle must be as close as practical to the steam condensing temperature. If superheat were introduced, the average temperature of heat addition for the bottoming cycle would drop, negating the efficiency gains usually derived from superheating.

In order to bring the steam and ammonia cycles together into one combined cycle certain modifications must be made to the two cycles. The modifications are:

(1) Replacing the water-cooled steam condenser with a steam condenser/ammonia boiler.

(2) Raising the backpressure of the low pressure steam turbine and combining the remaining LP turbine stages with the IP turbine.

(3) Removal of feedheating stages in the steam plant made unnecessary by raising the steam condensing temperature.

The T-S diagram and component flow chart for the combined cycle are shown as figures five and six respectively.

Since the high temperature end of the steam plant is not altered, the only independent parameter in the steam side of the combined plant is the steam condensing temperature, T_S. On the ammonia side there are two independent variables:

(1) The ammonia boiling temperature TF(1).

(2) The ammonia condensing temperature, TF(5).

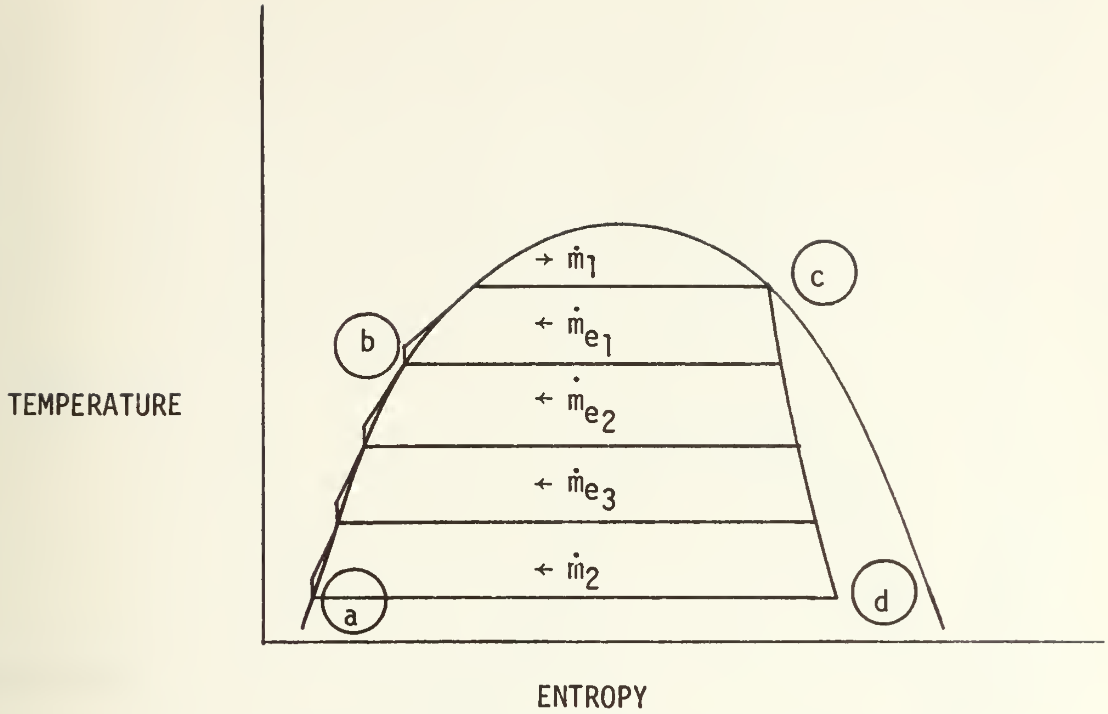


FIGURE 4 T-S DIAGRAM OF BOTTOMING CYCLE

STATE	DESCRIPTION
a	Condenser exit (saturated liquid)
b	Exit of third feedheater, boiler inlet (liquid)
c	Boiler exit, turbine inlet (saturated vapor)
d	Turbine exit, condenser inlet (liquid and vapor)
FLOW	DESCRIPTION
$\dot{m}_1$	Mass flow rate through boiler
$\dot{m}_2$	Mass flow rate through condenser
$\dot{m}_{e_1}$	Mass flow rate of feedheating extraction #1
$\dot{m}_{e_2}$	Mass flow rate of feedheating extraction #2
$\dot{m}_{e_3}$	Mass flow rate of feedheating extraction #3

TABLE 2 DESCRIPTION OF T-S DIAGRAM (FIGURE 4)

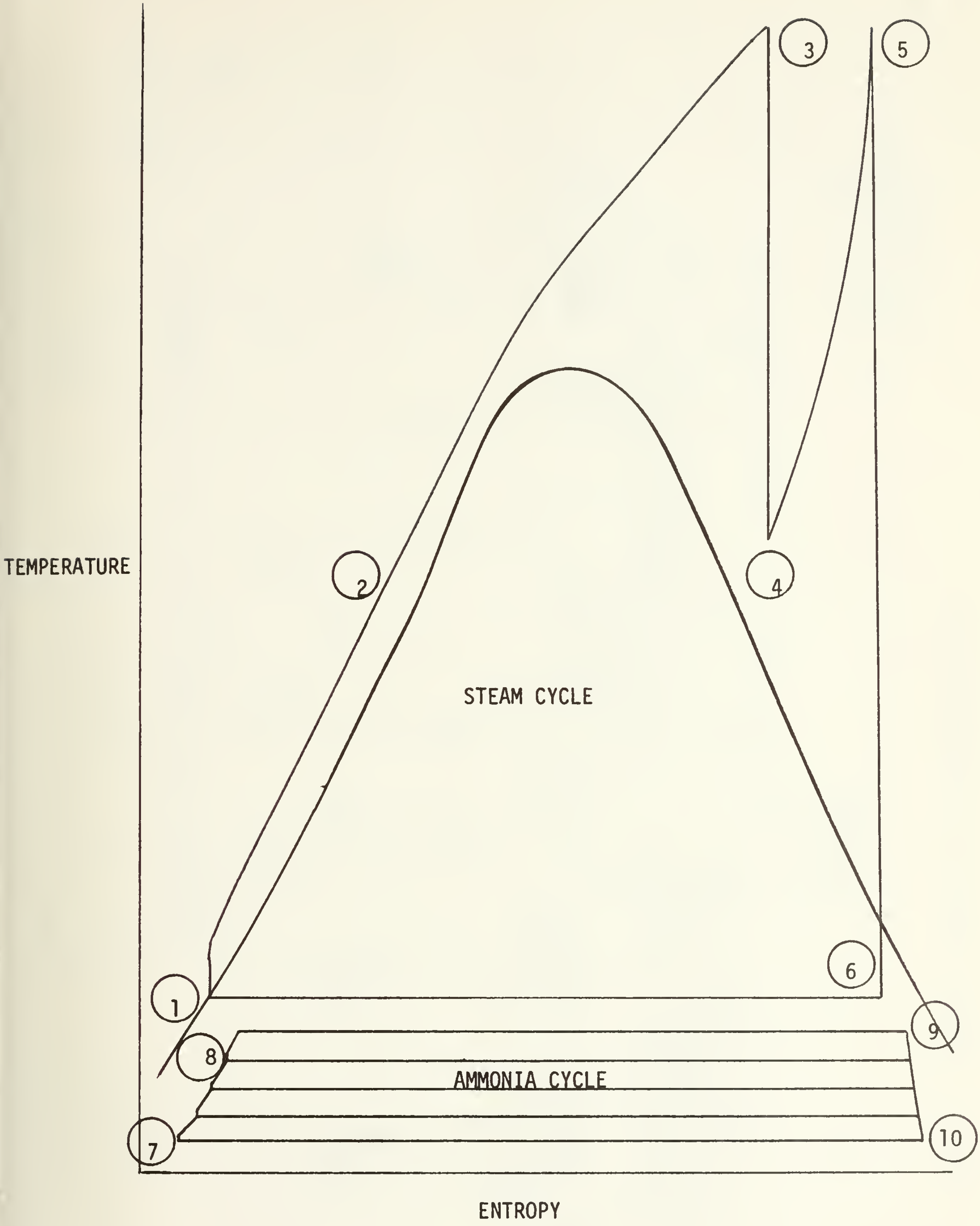


FIGURE 5 T-S DIAGRAM OF COMBINED CYCLE

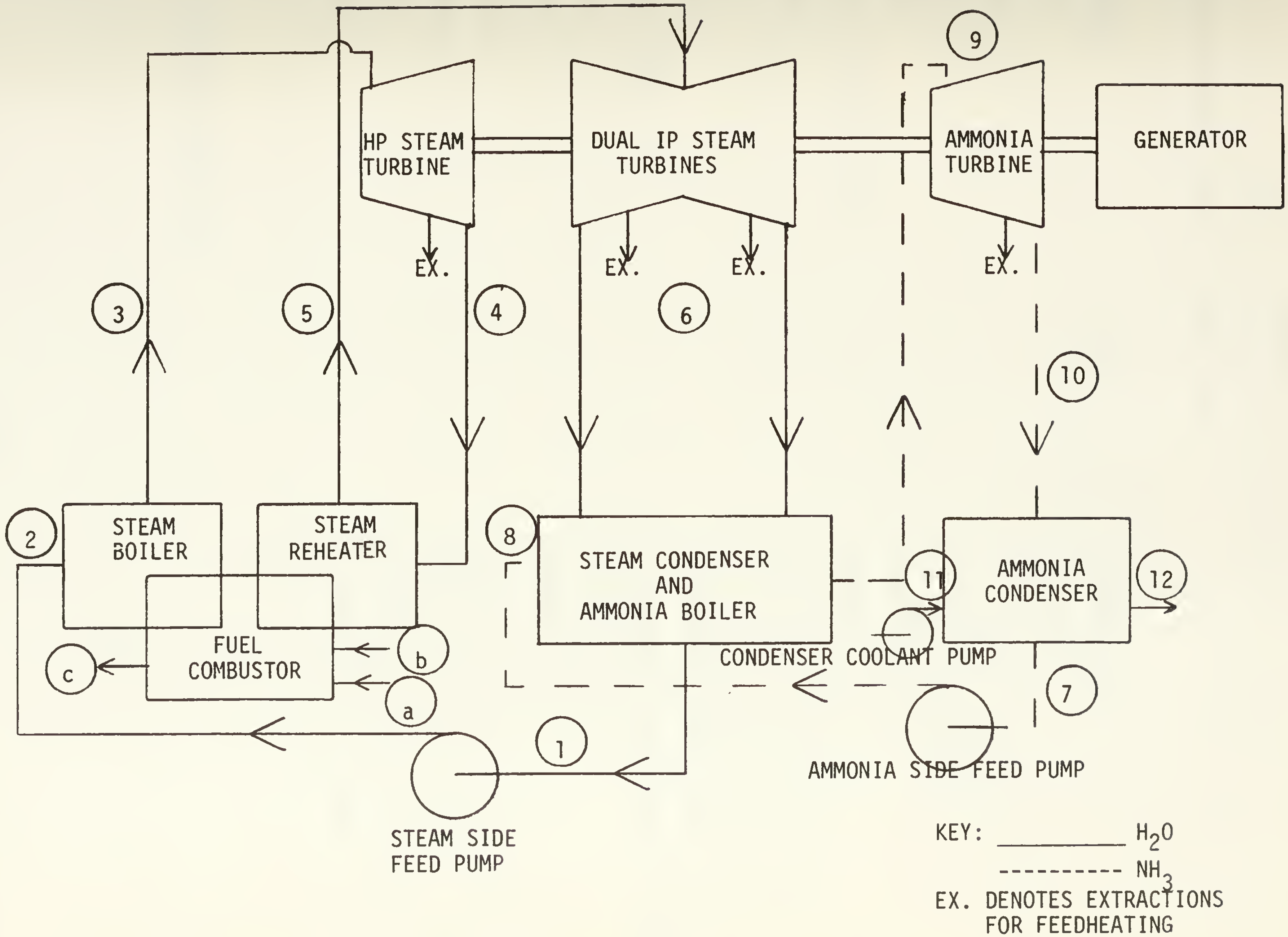


FIGURE 6 COMPONENT FLOW CHART FOR COMBINED CYCLE

TABLE 3: COMBINED CYCLE DESCRIPTION

STREAM NUMBERS REFER TO FIGURES 5 AND 6

STREAM	NO.	DESCRIPTION
Fuel	a	Fuel inlet
Air	b	Air inlet for combustion
Exhaust	c	Combustion products
Water	1	Condenser exit
Supercritical Liquid	2*	Boiler inlet
Steam	3	Boiler exit/HP turbine inlet
Steam	4	HP turbine exit/reheater inlet
Steam	5	Reheater exit/IP turbine inlet
Steam/Water	6	IP turbine exit/condenser inlet
Ammonia	7	Condenser exit
Ammonia	8*	Boiler inlet
Ammonia	9	Boiler exit/NH_3 turbine inlet
Ammonia	10	NH_3 turbine exit/condenser inlet
Water	11	Condenser coolant water inlet
Water	12	Condenser coolant water exit

* INCLUDES FEEDHEATING

The three variables mentioned above are the only independent variables in the analysis. The significant dependent variables are:

(1) The entropy averaged temperature of heat addition to the ammonia cycle, TAV, which is a function of TF(1) and TF(5).

(2) The difference between the steam condensing temperature T_S, and TAV. This variable is called $\Delta\bar{T}$ and is a function of all three independent variables. The magnitude of this parameter is related to the available energy lost due to the temperature drop between the cycles.

(3) The Log Mean Temperature Difference (LMTD) in the steam condenser/ammonia boiler. This factor which is numerically close to $\Delta\bar{T}$, is a function of the same three variables, and is a significant consideration in the sizing and cost estimation of the steam condenser/ammonia boiler.

The object of the combined cycle optimization is to consider how the three independent variables affect the plant performance and cost.

IV. OPTIMIZATION PRELIMINARIES

In this chapter the impact of the independent and dependent variables on the analysis will be examined. In addition, limitations on the range of these parameters will be discussed where appropriate.

a. The ammonia condenser and the ammonia condensing temperature.

In order to maximize the overall cycle efficiency, the lowest possible ammonia condensing temperature, TF(5), is desired. However, for a power plant with a water-cooled condenser there are three factors which limit how low the condensing temperature can be. The factors are:

(1) The condenser coolant water temperature at inlet must be greater than 32°F.

(2) The water velocity through the condenser must not be so high as to cause excessive corrosion in the tubes.

(3) The amount of power required to pump the coolant water through the condenser tubes must not negate the advantages of lowering the condensing temperature.

The reason why the water temperature at inlet must be greater than 32°F is that it must be a liquid in order to be pumped. If the ambient temperature is below 32°F, a cooling tower would be utilized in place of a wet condenser. The considerations of water velocity and pumping power are not quite so obvious, but are related to each other.

In a heat exchanger, such as a condenser, the temperature difference which will partially determine the condenser size is

$$\Delta T = T_C - (T_I + T_O)/2 \qquad \text{(equation 2)}$$

where T_C is the condensing temperature, T_I is the water inlet temperature,

and T_0 is the water exit temperature. The temperature vs. length plot for a condenser is shown figure 7.

The heat transferred in the condenser Q may be expressed as:

$$Q = UA\,\Delta T \qquad \text{(equation 3)}$$

where A is the heat transfer area and U is the overall heat transfer coefficient for the heat exchanger. The overall heat transfer coefficient may be computed by combining the thermal resistances in the unit as if they were resistors in a parallel electrical circuit. Assuming the thermal resistance of the condenser tubes to be negligible, and the thermal resistances for convection and condensation to be the inverse of the average heat transfer coefficients on their respective sides of the unit, then $U = 1/(1/h_{\substack{\text{condensing}\\ \text{fluid}}} + 1/h_{\substack{\text{convection on}\\ \text{water side}}})$ (equation 4)

For shorthand purposes the following symbolism will be used:

$h_{\text{condensing fluid}} \rightarrow h_{NH_3\ \text{cond}}$ and $h_{\text{convection on water side}} \rightarrow h_{\text{water}}$

In terms of the fluid and thermal parameters for a horizontal tube condenser:

$$h_{\substack{NH_3\\ \text{cond}}} = .728\ [1+.2C\ \Delta T(n-1)/h_{fg}]\ \sqrt[4]{(g(\rho)(\rho - \rho_v)k^3\ h'_{fg})/(nD\mu\ \Delta T_w)} \qquad \text{(equation 5)}$$

$$h_{\text{water}} = .023\ k_f/D\ (\rho_f VD/\mu_f)^{.8}\ (C_p\mu/k_f)^{.4} \qquad \text{(equation 6)}$$

where

C	=	specific heat of condensing liquid
C_p	=	specific heat of water
D	=	tube diameter
g	=	acceleration of gravity
h_{fg}	=	heat of vaporization of condensing fluid at condensing pressure
h'_{fg}	=	$1+(.68\ C\ \Delta T_w)/\ h_{fg}$
k	=	thermal conductivity of condensing liquid
k_f	=	thermal conductivity of water

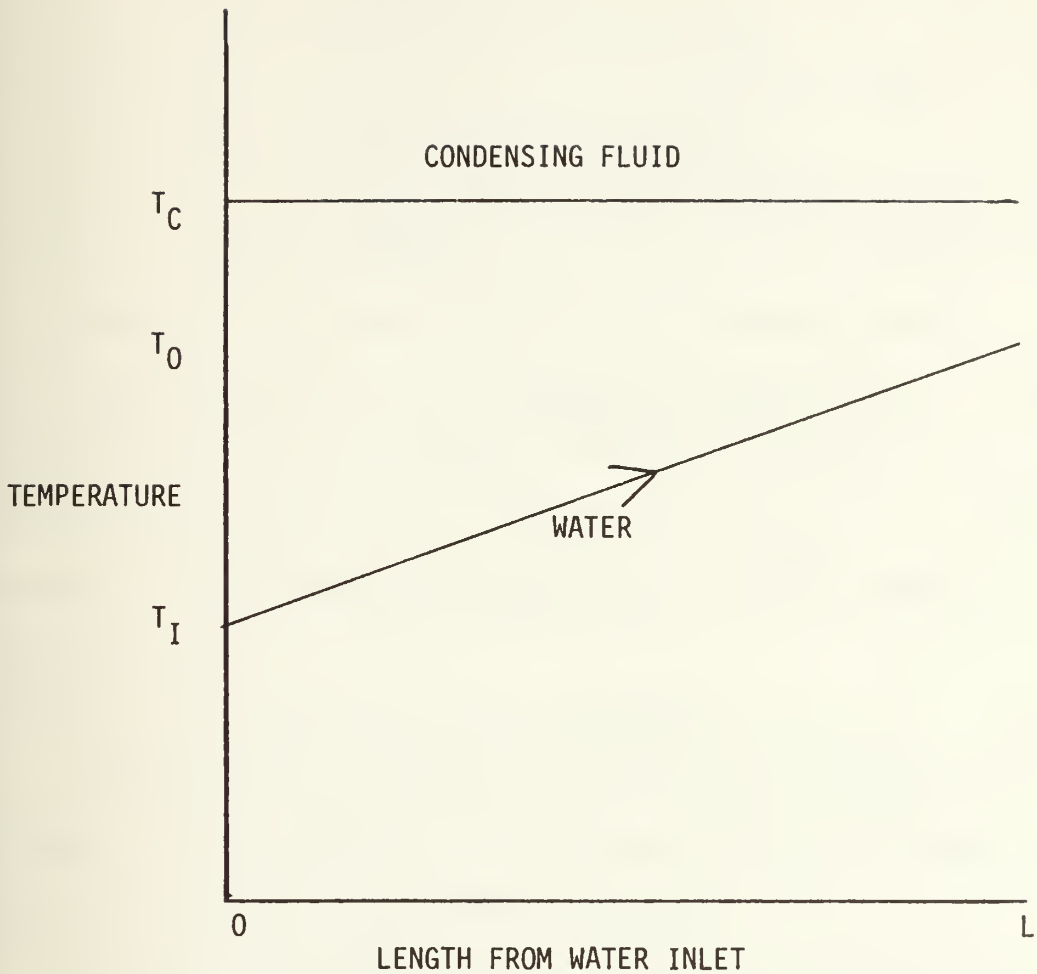


FIGURE 7 TEMPERATURE VS. LENGTH FOR A CONDENSER

n = number of tubes in vertical row

ρ = density of condensing liquid

ρ_f = density of water

ρ_v = density of condensing vapor

μ = dynanic viscosity of condensing liquid

μ_f = dynamic viscosity of water

V = water velocity through the tubes

ΔT_w = temperature difference between condensing fluid and tube wall

Furthermore, an energy balance for the condenser states:

$$Q = \dot{m}_{water} C_{p_{water}} (T_0 - T_I) = \dot{m}_{condensing\ fluid}\ x_e\ h_{fg} \qquad \text{(equation 7)}$$

and solving for $T_0 - T_I$, the water temperature increase:

$$T_0 - T_I = (\dot{m}_{condensing\ fluid}\ x_e\ h_{fg})/(\dot{m}_{water} C_{p_{water}}) \qquad \text{(equation 8)}$$

where x_e is the condensing fluid quality. Combining equations (2), (7), and (8) then,

$$\Delta T = T_C - (T_I + T_0)/2 = T_C - (T_I + Q/2\dot{m}_{water}) \qquad \text{(equation 9)}$$

From equation (3) it may be seen that for a condenser to be of minimum size and cost once the heat transfer amount Q has been selected, the product $U\Delta T$ must be as great as possible.

However, from equations (5) and (6), it may be shown that ΔT changes much more rapidly than U, apparently stating that ΔT should be maximized. Equation (9) indicates that once T_C is chosen (and ideally as low as possible), there are only two ways to increase ΔT. The first method is to decrease T_I, the water inlet temperature, which is difficult since T_I is a function of the plant site.

On the other hand, if the $Q/2\dot{m}_{water}$ term in equation (9) is small the value of ΔT will remain large. The problem is that the only way to

decrease the $Q/2\dot{m}_{water}$ term for a fixed Q, is to increase the mass flow rate of water through the condenser tubes. This brings one back to the problems of water velocity and pumping power stated earlier in this chapter.

In order to keep the economic analysis of the components as simple as possible, the combined cycle plant condenser under consideration will have the same ratio of cycle power output to surface area as the 'Bull Run' unit. This assumption is valid, since the condenser limitations are on the water side. The 'Bull Run' condensing system consists of 41368 tubes, each of which has an outer diameter of .875", and a total heat transfer surface area of 320,000 $feet^2$. Since the mass flow rate, $\dot{m}_{water} = \rho_{water} V_{water} A_{flow}$ (equation 10)

and $A_{flow} = n\pi D^2/4$ (equation 11)

(where A_{flow} is the area perpendicular to the direction of water flow, n is the number of tubes, and D is the tube inner diameter), then

$V_{water} = \dot{m}_{water}/\rho_{water} A_{flow} = 4\dot{m}_{water}/\rho_{water} n\pi D^2$ (equation 12)

Now that an expression for the water velocity as a function of the mass flow rate and the condenser geometry has been defined, a limit on the mass flow rate and TF(5) can be specified once a limitation on the water velocity is established.

Since ammonia reacts readily with copper, stainless steel is the likely candidate for the tube material. Studies indicate that for steel tubes corrosion will not be excessive as long as the water velocity is below 15 ft/sec.[11]

This water velocity may require excessive pumping power which would be counter-productive. The pumping power, PP may be expressed as: $PP = (\Delta P \times Q)/\eta_p$ (equation 13)

where Q is the volumetric flow rate, η_p the pump efficiency, and ΔP is the pressure drop through the tubes.

The volumetric flow rate Q, is

$Q = \dot{m}_{water}/\rho_{water}$ (equation 14)

and the pressure drop, ΔP is:

$\Delta P = 4\, f(L/D)(\rho_{water} V^2_{water}/2g_o)$ (equation 15)

where f is the pipe friction factor, L the length and g_o is the units correction factor.

For turbulent flow in tubes:

$f = .079/R_e^{.25}$ (equation 16)

where R_e is the Reynolds number and is equal to

$R_e = (\rho_{water} V_{water} D)/\mu_{water}$ (equation 17)

Combining equations (13) through (17), the total pumping power is:

$PP = .316 / (\rho_w V_w D/\mu_w)^{.25} (L/D)(\rho_w V_w^2 /2g_o) (\dot{m}_w/\rho_w)(1/\eta_p)$ (eq. 18)

recalling that L = TOTAL HEAT TRANSFER AREA (A)/$(n\pi D)$ (eq. 19)

and $\dot{m} = (\rho_w V_w\, n\pi D^2)/4$ (equation 20)

the pumping power expression simplifies to:

$PP = (.079\, A\, \mu^{.25} \rho^{.75} V^{2.75})/ (4\, D^{.25} g_o\, \eta_p)$ (equation 21)

in which V is the most significant factor and the only one that will change noticeably from one set of parameters to another.

Thus far it has been shown how the ambient water temperature T_I, and practical engineering considerations limit the ammonia condensing temperature.

To summarize:

(1) Once a plant site has been chosen T_I is fixed.

(2) For a fixed T_I and a given condenser, the mass flow rate (and velocity) must be maximized in order to keep the ammonia condensing temperature as low as possible.

(3) Limitations on the water velocity are imposed by corrosion considerations.

(4) In addition to #3, the water velocity must not be so large as to require excessive pumping power which would negate the advantages of lowering TF(5), the NH_3 condensing temperature.

The interface between the gains of lowering TF(5) and the pumping power requirements will be considered later.

b. Steam Condensing Temperature, T_S

It is desired to find the steam condensing temperature which will give the maximim cycle efficiency. To obtain the optimum T_S, it must be determined whether the steam or ammonia cycle makes better use of the low temperature end of the cycle.

Assuming that the steam and ammonia turbines have the same isentropic efficiencies in the two-phase region, the deciding factor in the selection of T_S, is the power loss due to moisture droplets in the turbine.

Common practice indicates that the power loss due to wetness is

one percent for each percent of average wetness in that turbine stage.

Applying this wetness consideration to the 'Bull Run' plant, a plot of power loss due to wetness versus the steam condensing temperature may be created as shown in figure 8.

Once the ammonia condensing temperature TF(5) has been selected, the power loss analysis may be carried out for the ammonia cycle as well. However, since the third independent variable, the NH_3 boiling temperature TF(1), has not been fixed the wetness calculation must be carried out for each value of TF(1). This procedure produces a family of curves one of which is shown in figure 9.

Figure 8 demonstrates that the power loss for the steam cycle increases as T_S drops, and figure 9 shows the opposite trend for the ammonia cycle.

The optimum value of T_S may be found by superimposing figures 8 and 9 and plotting the combined cycle wetness loss versus T_S.

Figure 10 shows the superimposition of figures 8 and 9 and the resulting total wetness loss. Note that a mimimum loss occurs around $T_S = 131^{\circ}F$ for the parameters chosen. Furthermore, if the combined cycle efficiency is plotted versus the non-dimensionalized temperature difference between the steam and ammonia cycle for various values of T_S, as shown in figure 11, it is clear that $T_S = 131^{\circ}F$ is the optimum choice.

The selection of a steam condensing temperature at this level is different from the studies cited in chapter II. All of the earlier work

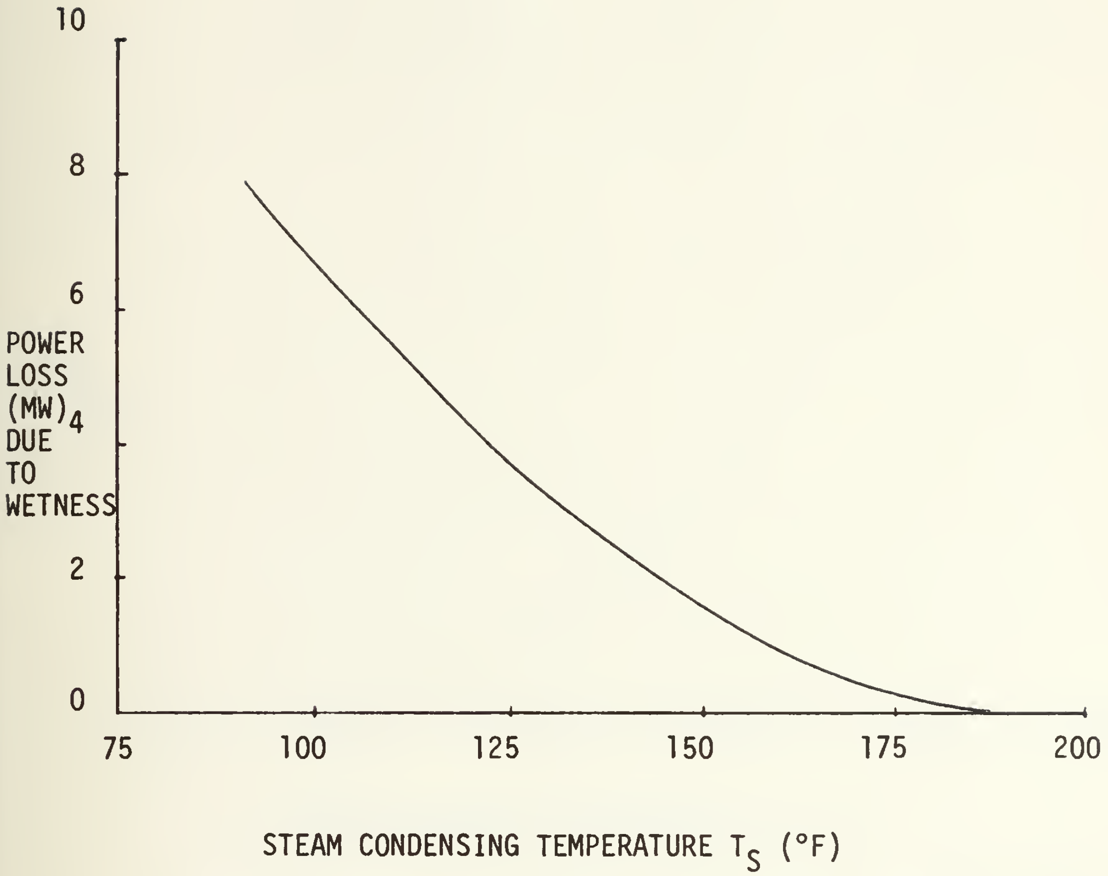


FIGURE 8 WETNESS POWER LOSS VERSUS STEAM CONDENSING TEMPERATURE T_S FOR STEAM CYCLE

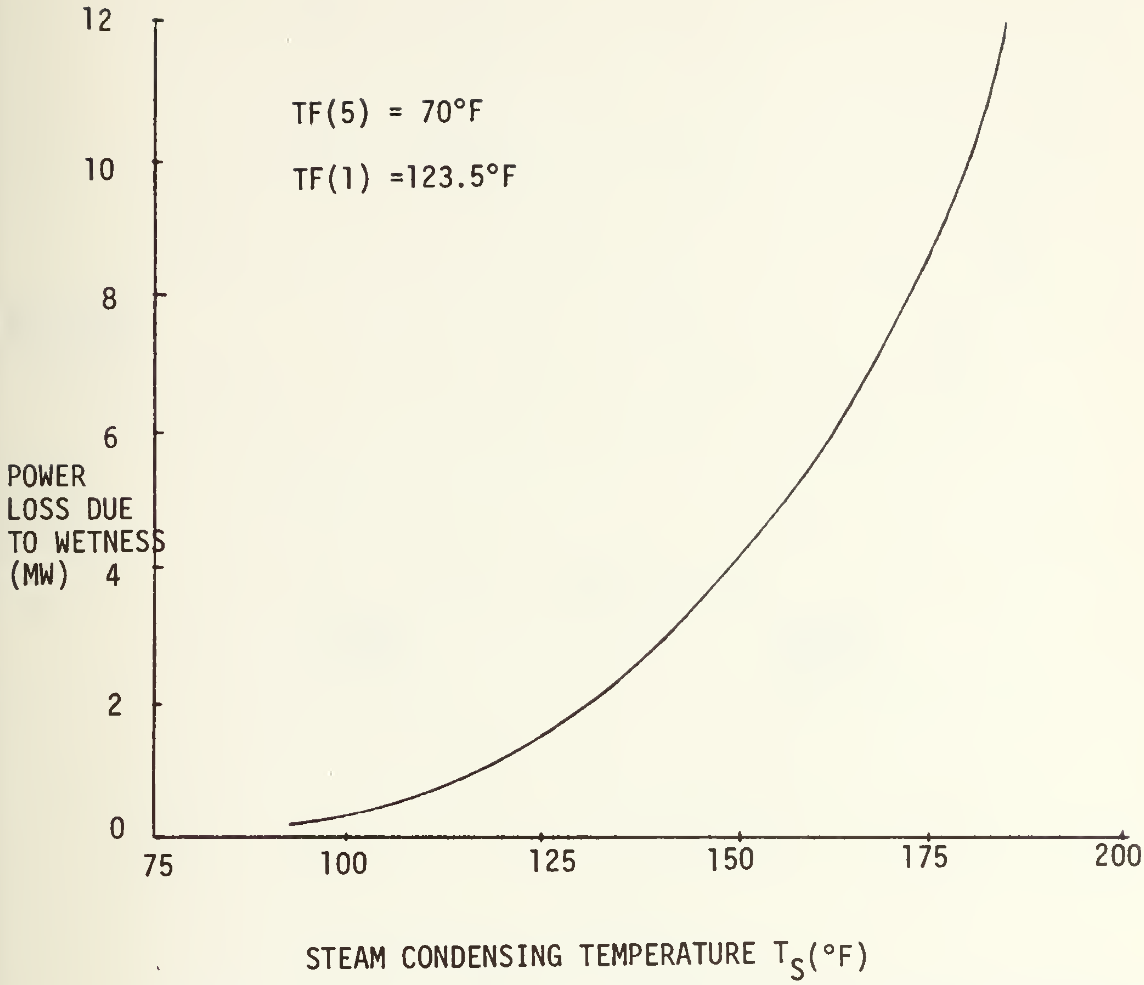


FIGURE 9 WETNESS POWER LOSS VERSUS STEAM CONDENSING TEMPERATURE T_S FOR NH_3 CYCLE

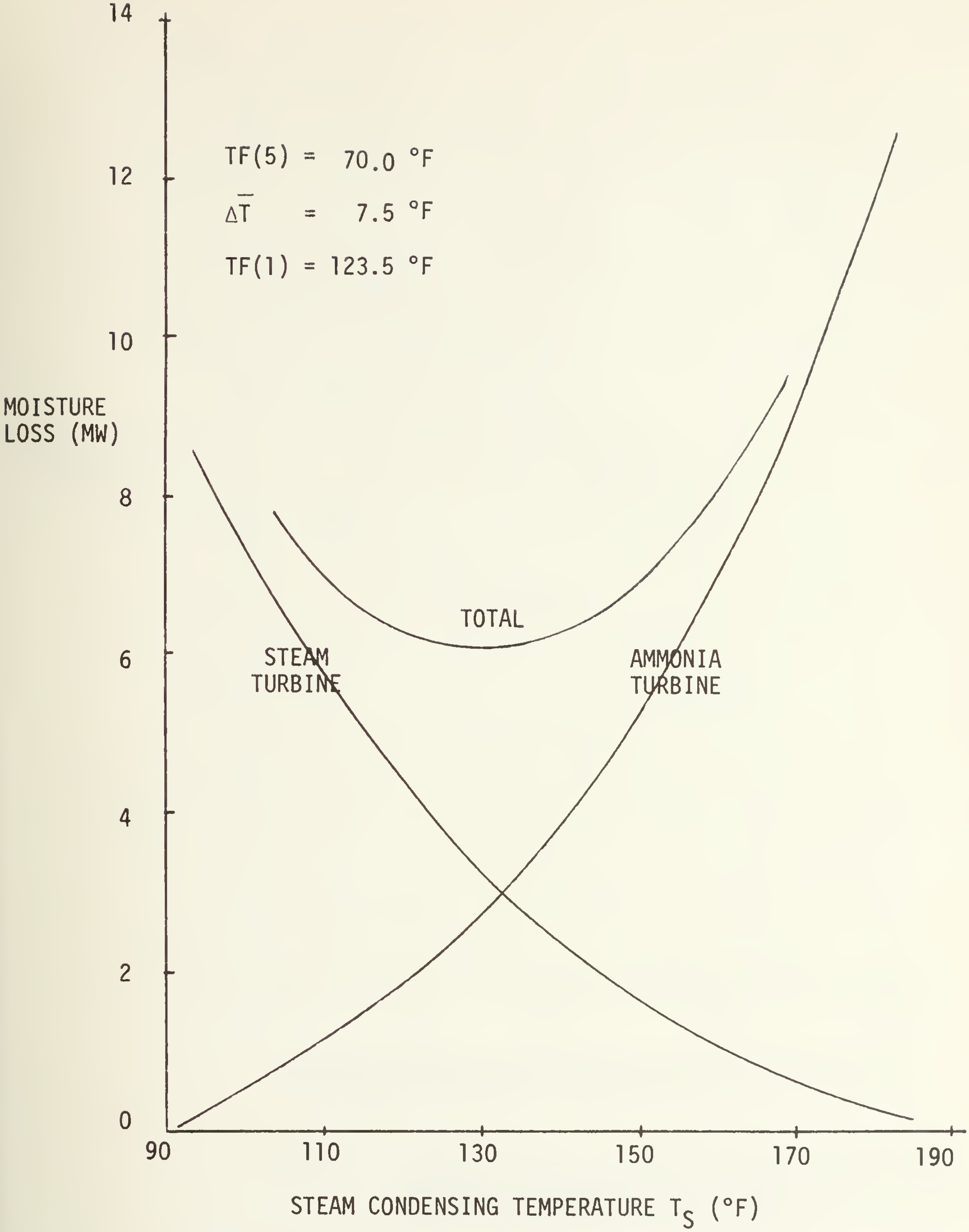


FIGURE 10 MOISTURE LOSS VERSUS STEAM CONDENSING TEMPERATURE

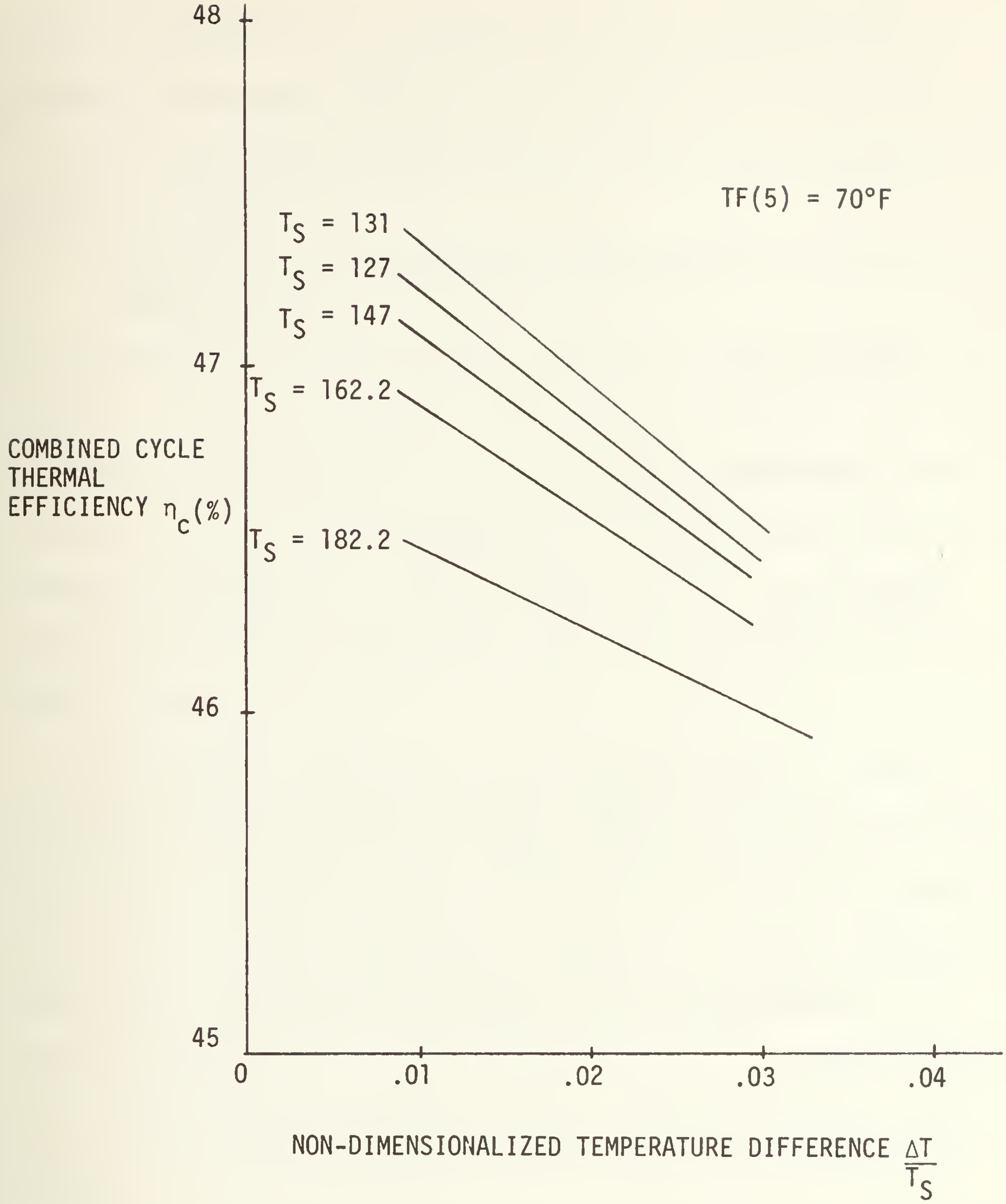


FIGURE 11 COMBINED CYCLE EFFICIENCY VERSUS NON-DIMENSIONALIZED TEMPERATURE DIFFERENCE FOR VARIOUS STEAM CONDENSING TEMPERATURES

assumed a value of T_S in the neighborhood of 220° - 250°F without considering the wetness losses.

The analysis which generated figures 10 and 11 will be discussed in the next chapter.

c. The Ammonia Boiling Temperature TF(1) and the Steam Condenser/ Ammonia Boiler.

The third independent variable is TF(1), the ammonia boiling temperature. This variable is most significant in its impact on the cycle efficiency and the size and cost of the steam condenser/ammonia boiler. The value of TF(1) is not as important as its relationship with the steam condensing temperature T_S. This relationship defines the dependent variable $\Delta\overline{T}$, the average temperature difference between the steam and ammonia cycles.

As indicated in chapter three and graphically shown in figure 11, the combined cycle efficiency is highest when $\Delta\overline{T}$ is as small as possible. Unfortunately, since the average temperature difference $\Delta\overline{T}$ is numerically similar to the log mean temperature difference in the steam condenser/ ammonia boiler and the heat transferred in the heat exchanger Q is:

$$Q = UA\ \Delta T_{\text{Log mean}}. \qquad \text{(equation 22)}$$

Thus as the temperature difference decreases the heat exchanger area increases.

It is obvious that the efficiency gains derived by a small $\Delta\overline{T}$ may be negated by the additional capital cost of a large steam condenser/ ammonia boiler.

The task at hand is twofold:

(1) Determine how $\Delta\overline{T}$ affects the overall thermal efficiency.

(2) Determine how $\Delta\overline{T}$ affects the steam condenser/ammonia boiler size and cost.

Task one will be considered in the next chapter.

In order to determine how $\Delta\overline{T}$ affects the heat exchanger size and cost, a detailed heat transfer analysis of the unit must be made.

The steam condenser/ammonia boiler to be considered is a shell and tube arrangement with the steam condensing outside the array of horizontal tubes through which the ammonia flows.

For analytical purposes the SC/AB may be divided into two zones. In the first zone, the ammonia is heated from the feedheating outlet temperature, TF(2) to the ammonia boiling temperature TF(1). The heat transfer mechanism on the ammonia side in this region is forced convection, with subcooled boiling effects being neglected, and on the steam side the mechanism is condensation.

In the second zone, the ammonia temperature remains constant as the ammonia is heated from a saturated liquid to a saturated vapor state. The heat transfer mechanism on the NH_3 side in this zone is very complex, and includes effects of nucleate boiling and forced convection. The heat transfer mechanism on the steam side is once again condensation.

In both the non-boiling (zone 1) and boiling (zone 2) regions the heat transfer coefficient on the steam side h_{steam} may be expressed as:

$$h_{steam} = .728\ [1+.2C\ \Delta T(n-1)/h_{fg}]\ \sqrt[4]{(g(\rho)(\rho-\rho_v)k^3\ h'_{fg})/nD\mu\ \Delta T_w)} \quad \text{(eq. 23)}$$

where the equation and symbols are identical to equation 5, which was used

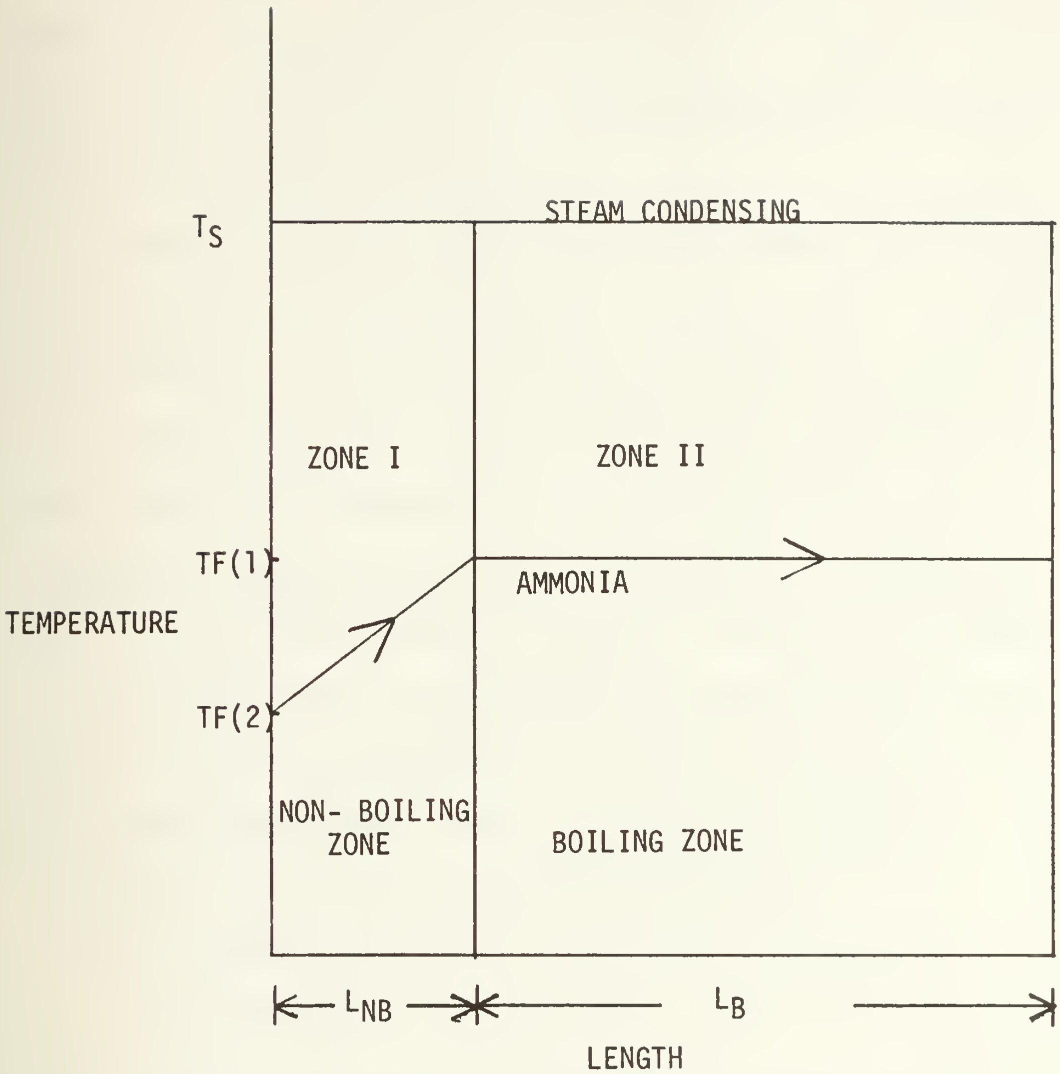


FIGURE 12 TEMPERATURE VERSUS LENGTH FOR BOTH ZONES OF THE STEAM CONDENSER/AMMONIA BOILER

in the NH_3 condenser calculations.

On the ammonia side in the non-boiling region the forced convection coefficient is: $h_{NH_3\ FC} = .023\ (k_f/D)\ (\rho_f VD/\mu)^{.8}\ (C_p\mu/k_f)^{.4}$ (eq. 24) which is the same expression as equation(6),except that the fluid properties for ammonia are used here.

In zone 2, the boiling region, the heat transfer coefficient may be calculated by using several empirical relationships. The most reliable seems to be the Chen correlation[12] which states that the heat transfer coefficient in the two-phase region where boiling and forced convection are present, may be calculated as follows:

$h_{TWO\ PHASE} = h_{NUCLEATE\ BOILING} \times S + h_{FORCED\ CONVECTION\ 2\ PHASE} \times F$ (eq. 25)

where S is the boiling suppression factor, a measure of how much the fluid motion upsets the normal boiling mechanism and F is the Chen factor which is a measure of how much the fluid motion induced by boiling upsets the normal forced convection mechanism.

The Chen factor may be calculated as follows:

(1) Determine the Martinelli-Nelson factor X_{tt}, where

$$X_{tt} \cong (\rho_g/\rho_f)^{.5}\ (\mu_f/\mu_g)^{.1}\ ((1-x)/x)^{.9} \qquad \text{(equation 26)}$$

Note that ρ_g and ρ_f represent the density of saturated vapor and liquid respectively, μ_f and μ_g the viscosity of saturated liquid and vapor respectively, and x is the thermodynamic quality.

(2) Utilizing figure 13, which displays F versus $1/X_{tt}$, read off F.

The boiling suppression factor S may be found as follows:

(1) Find the two-phase Reynolds number $R_{e_{TP}}$ which is represented by

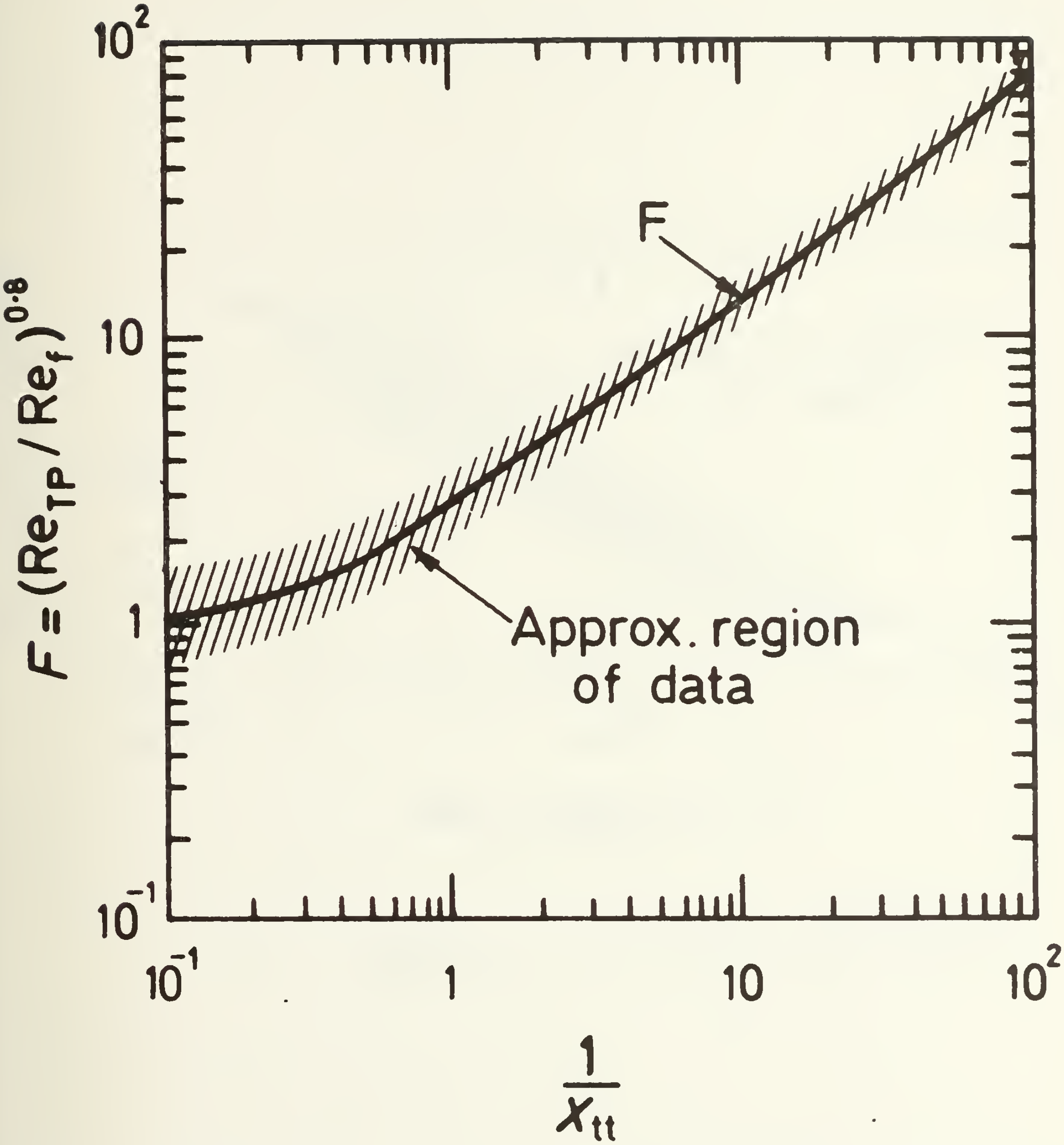


FIGURE 13 CHEN FACTOR VERSUS INVERSE OF MARTINELLI-NELSON FACTOR[13]

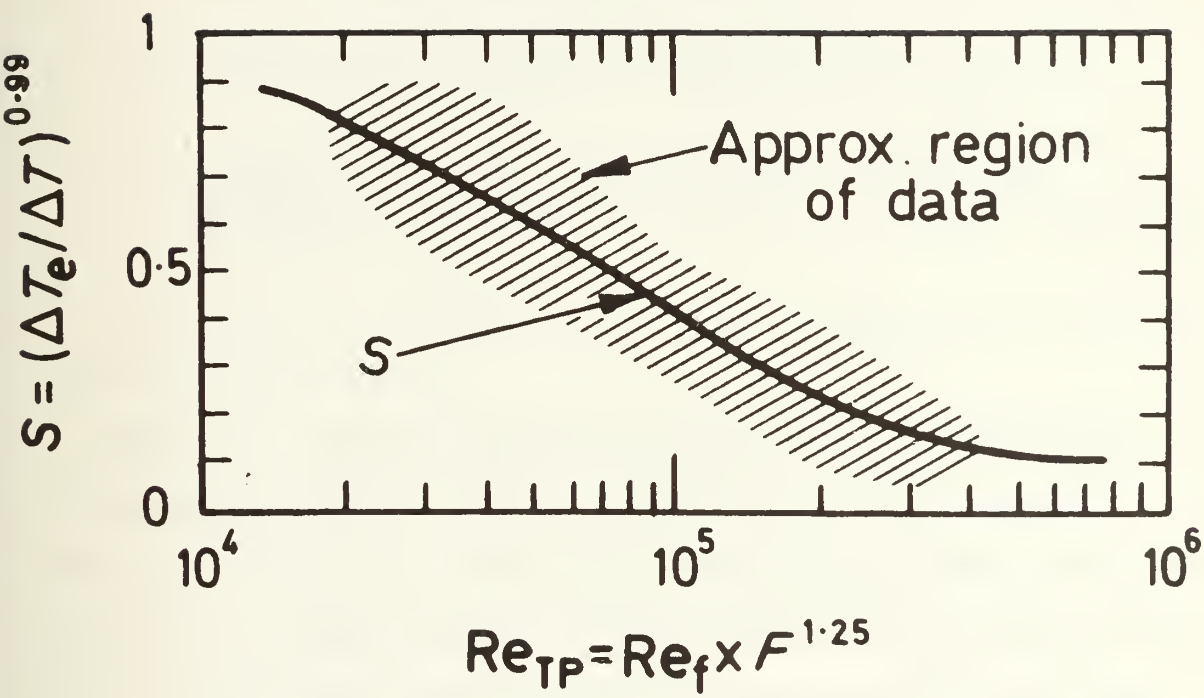


FIGURE 14 SUPPRESSION FACTOR VERSUS TWO-PHASE REYNOLDS NUMBER[14]

(3) Utilizing steps one and two the forced convection of ammonia heat transfer coefficient may be found.

(4) Solve the heat flux equation by

$$Q_{NB}/A_{NB} = h_{steam}\ (T_S-T_{wall}) = h_{NH_3\ FC}\ (T_{wall}-T_{NH_3}) \quad \text{(eq. 33)}$$

iteration. The reason that this relationship cannot be solved in closed form is that the expression for h_{steam} (equation 23) contains the term $(T_S-T_{wall})^{-1/4}$, making a numerical result preferable. The iteration is carried out by assuming a tube wall temperature and incrementally changing T_{wall} until h_{steam} (T_S-T_{wall}) equals $h_{NH_{3FC}}(T_{wall}-T_{NH_3})$. Note that T_{NH_3} is the numerical average between the ammonia inlet and outlet temperature.

(5) Divide the non-boiling heat transfer by the heat flux to determine the area of the non-boiling section, A_{NB}.

(6) Calculate the pressure drop in this section of the SC/AB and determine that the design choices made in step one are responsible. Use equation 15 and substitute ammonia properties for those of water.

The procedure for the boiling section calculations are basically the same, but are more complex.

Note that in equation(26)and(30),the variable 'x' which represents the thermodynamic quality appears. In order to calculate the heat transfer coefficient for a region where the quality goes from zero to one, what value of x should be used? A simple average between the end values is not suitable since in both equations(26)and(30),the 'x' term is non-linear. However, in order to make this problem less difficult to solve, an approximation which yields good accuracy is to use an average quality in the analysis, provided the change of the quality is small.

In keeping with this idea, the boiling region of the SC/AB is divided into ten segments, each having a quality change of one-tenth and

and utilizing the average quality in that segment in equations (26) and (30).

The configuration of the steam condenser/ammonia boiler is determined (step one for non-boiling section) and the amount of heat transfer in the boiling region, Q_B, as well as the NH_3 mass flow rate (step 2) are fixed. Then for each quality segment of the boiling region the heat flux iteration must be performed:

$$\frac{Q_{B\ SEGMENT}}{A_{B\ SEGMENT}} = h_{steam}(T_S - T_{wall}) = (h_{\substack{NUCL\\BOIL}} S + h_{\substack{2PHASE\\FORC.\ CONV}} F)(TF(1) - T_{wall}) \quad \text{(eq. 34)}$$

This iteration is more complex than the non-boiling case since $h_{\substack{NUCLEATE\\BOILING}}$ and h_{steam} both contain temperature difference terms raised to fractional powers.

Once the heat flux for each segment has been determined, the heat transfer area of that segment may be found. The total boiling region area A_B is the total of the areas of the ten quality segments.

Finally, the pressure drop in the boiling region must be evaluated to determine design feasibility. Using a homogeneous flow model[16], the pressure drop for the boiling region is:

$$\Delta P = \frac{2f_{TP}\ \rho V^2 L}{D} \quad 1 + \frac{v_{fg}}{2v_f} + \rho V^2\ (v_{fg}/v_f) \qquad \text{(equation 35)}$$

where $f_{TP} = .079/\ (\rho VD/\bar{\mu})^{.25}$ (equation 36)

and $\bar{\mu}$ is the average viscosity and may be expressed by the McAdams correlation[17]: $\bar{\mu} = x\mu_g + (1-x)\mu_f$. (equation 37)

Once again quality is important and the calculation of the pressure drop is carried out in the same manner as the heat transfer coefficients, by assuming the calculation to be piecewise linear.

d. Chapter summary

In this chapter, the impact of the three independent variables TF(5), TF(1), and T_S on the heat exhanger design and cycle efficiency have been examined. Specifically, the NH_3 condenser performance and pumping requirements, the selection of an optimum T_S, and the sizing of the steam condenser/ammonia boiler have been considered. In the next chapter, the thermodynamic analysis will be discussed and the impact of the three variables on cycle efficiency.

V. THERMODYNAMIC ANALYSIS OF THE COMBINED CYCLE

In this chapter, the impact of the three design parameters, the ammonia boiling temperature, ammonia condensing temperature, and steam condensing temperatures on the combined cycle efficiency will be studied.

a. The Steam Cycle

The steam condensing temperature, T_S, is the only variable which affects the thermal efficiency of the steam cycle. Specifically, it would be desirable to know how the steam cycle efficiency η_{ST}, varies with T_S, the steam condensing temperature.

The power output of the steam cycle is affected by two considerations as the steam condensing temperature is increased to accomodate the bottoming cycle. The first of these considerations is that the LP steam turbine produces less power due to the higher exhaust backpressure. Secondly, since the steam is being condensed at a higher temperature than the 'Bull Run' plant, less steam must be extracted from the power producing turbines for feedheating purposes. Note that those considerations affect the cycle efficiency in opposing manners.

With regard to the loss of power due to increased backpressure, only the low pressure steam turbine is affected. The power loss associated with the higher backpressure is: $PL = \dot{m}_{steam} (h_i - h_o)$ (equation 38) where h_i is the enthalpy of the stage exhaust steam for the increased back-pressure case and h_o is the stage exhaust enthalpy for the unmodified Bull Run plant. Equation (38) must be modified to include the difference

in wetness losses between the original and the modified steam plant. Accordingly if h_i is the inlet enthalpy, $\bar{x}_1$ is the average quality for the modified plant, and $\bar{x}_0$ is the average quality for the base case, equation (38) becomes:

$$PL = \dot{m}_{steam} \left((h_i - h_o) \bar{x}_0 - (h_i - h_1) \bar{x}_1 \right) \qquad \text{(equation 39)}$$

Equation (39) is valid for each stage in the LP turbine provided that the proper steam flow rate, $\dot{m}_{steam}$ is used.

In addition to the difference in power output due to the higher backpressure, the changes in the feedheating extractions also affect the power output.

In the 'Bull Run' plant, the extracted steam is condensed in a closed-type feedheater, transferring heat to the feedwater until the condensed extraction temperature is 5°F higher than the feedwater outlet temperature. Ultimately, the condensed extraction is mixed with the feedwater from the main steam condenser.

In order to determine the cycle power variation due to changes in feedheating extractions, an expression for the mass flow rate of steam 'a' in figure 15 must be established.

Referring to figure 15, an energy balance of the closed feedheater indicates:

$$\dot{m}_a (h_a - h_b) = \dot{m}_d (h_e - h_d) \qquad \text{(equation 40)}$$

A design constraint for the 'Bull Run' feedheaters states:

$$T_b = T_e + 5°F \qquad \text{(equation 41)}$$

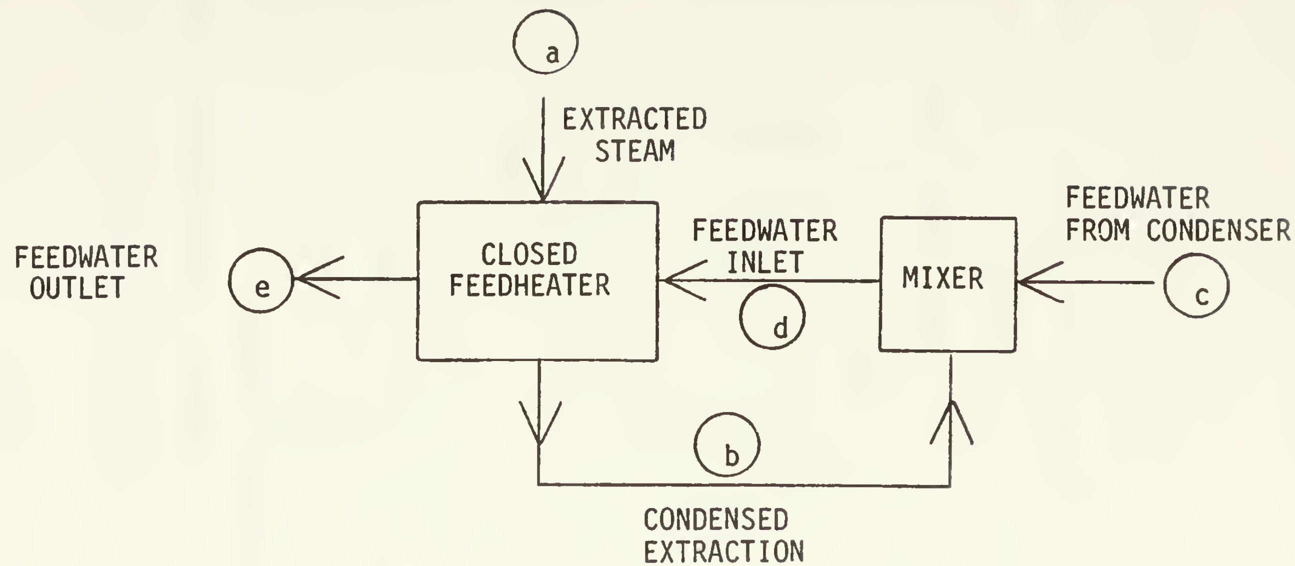


FIGURE 15 SIMPLIFIED FEEDHEATER FLOW DIAGRAM

and since $C_{p_{water}} = 1$ BTU/lbm - °R, then:

$$h_b = h_e + 5 \text{ BTU/lbm} \qquad \text{(equation 42)}$$

An energy and mass balance in the mixer states:

$$\dot{m}_d = \dot{m}_a + \dot{m}_c \qquad \text{(equation 43)}$$

$$h_d = (\dot{m}_a h_b + \dot{m}_c h_c)/(\dot{m}_a + \dot{m}_c) \qquad \text{(equation 44)}$$

Combining equations (40) through (44), the extraction flow, $\dot{m}_a$, may be expressed as:

$$\dot{m}_a = \dot{m}_c \left[\frac{(\dot{m}_a + \dot{m}_c)h_e - \dot{m}_a(h_e + 5) - \dot{m}_c h_c}{(\dot{m}_a + \dot{m}_c)(h_a - 2h_e - 5) + \dot{m}_a(h_e + 5) + \dot{m}_c h_c}\right] \qquad \text{(equation 45)}$$

The condenser flow rate is also affected by the amount of feedheating extraction. The definitive relationship is:

$$\dot{m}_c = \dot{m}_{BOILER} - \dot{m}_{OTHER\ EXTRACTIONS} - \dot{m}_a \qquad \text{(equation 46)}$$

renaming $\dot{m}_{BOILER} - \dot{m}_{OTHER\ EXTRACTIONS} = \dot{m}_g$ (equation 47)

and combining equation (45), (46), and (47):

$$\dot{m}_a = (\dot{m}_g - \dot{m}_a)\left[\frac{\dot{m}_g h_e - \dot{m}_a(h_e + 5) - (\dot{m}_g - \dot{m}_a)h_c}{\dot{m}_g(h_a - 2h_e - 5) + \dot{m}_a(h_e + 5) + (\dot{m}_g - \dot{m}_a)h_c}\right] \qquad \text{(eq. 48)}$$

The variables in equation (48), $\dot{m}_g$ and h_a are fixed at the values of the unmodified plant.

The feedheater exit enthalpy will be fixed as a design constraint in this analysis. Thus, there are only two unspecified variables, the first is the enthalpy of the main condenser condensate, h_c, which is a direct function of the steam condensing temperature, T_S. The second variable is the extraction mass flow rate, $\dot{m}_a$. Based on the arguments presented, it is now possible to state that: $\dot{m}_a = f(T_S)$ (eq. 49)

where the function f is specified in equation (48) and in the relationship between h_c and T_S, which may be found in the steam tables.

Therefore, if the extraction mass flow rate for the unmodified cycle is $\dot{m}_{a_0}$, and the extraction flow rate for the new steam condensing temperature is $\dot{m}_{a_1}$, the power loss due to the higher condensing temperature (including the effects of equation (39)) is:

$$PL = \dot{m}_{s_0}(h_i - h_0)\overline{x}_0 - \dot{m}_{s_1}(h_i - h_1)\overline{x}_1 \qquad \text{(equation 50)}$$

where $\dot{m}_{s_0}$ is the turbine stage mass flow rate for the unmodified case, and $\dot{m}_{s_1}$ is the modified cycle mass flow rate. The modified cycle flow rate may be expressed as:

$$\dot{m}_{s_1} = \dot{m}_{s_0} + (\dot{m}_{a_0} - \dot{m}_{a_1}) \qquad \text{(equation 51)}$$

Combining equations (50) and (51), the power loss is :

$$PL = \dot{m}_{s_0}(h_i - h_0)\,\overline{x}_0 - (\dot{m}_{s_0} - (\dot{m}_{a_0} - \dot{m}_{a_1}))(h_i - h_1)\,\overline{x}, \qquad \text{(eq. 52)}$$

The thermal efficiency of the modified steam cycle becomes:

$$\eta_{th_{steam}} = (P_0 - PL)/Q_{A_{steam}} \qquad \text{(equation 53)}$$

where P_0 is the unmodified cycle power output and $Q_{A_{steam}}$ is the heat added to the steam cycle.

Consequently, the heat rejected by the steam cycle, which is also the heat added to the bottoming cycle is:

$$Q_{R_{steam}} = Q_{A_{NH_3}} = Q_{A_{steam}}(1 - \eta_{th_{steam}}) \qquad \text{(equation 54)}$$

It may be noted that effect of only one stage of feedheating was shown in the modification of the 'Bull Run' plant. For the range of values where the steam condensing temperature T_S, yields optimum combined

cycle efficiency, only the first feedheating stage is affected. However, a more complex analysis was undertaken for values of T_S which affected the second and third feedheating stages.

b. The Ammonia Bottoming cycle

The ammonia bottoming cycle is a Rankine cycle with three stages of feedheating and no superheating.

In carrying out the thermodynamic analysis of the bottoming cycle, the following symbols are used:

s = entropy (BTU/lbm °R)

h = enthalpy (BTU/lbm)

P = Pressure (psi)

TF = temperature (°F)

v = specific volume (ft^3/lbm)

f = used as a subscript, indicates saturated liquid state

g = used as a subscript, indicates saturated vapor state

fg = used as a subscript, indicates property change associated with vaporization or condensation

Figure 16 is the T-S diagram for the bottoming cycle with the nomenclature to be used in the analysis. Note that since there are three stages of feedheating, there are five temperature levels in the cycle. The temperature levels are numbered sequentially, starting with level one which is the ammonia boiling temperature and culminating with level five which is the ammonia condensing temperature.

The power extracted from each turbine stage is the product of the mass flow rate and the enthalpy drop in the stage, minus mechanical, kinetic energy and moisture losses.

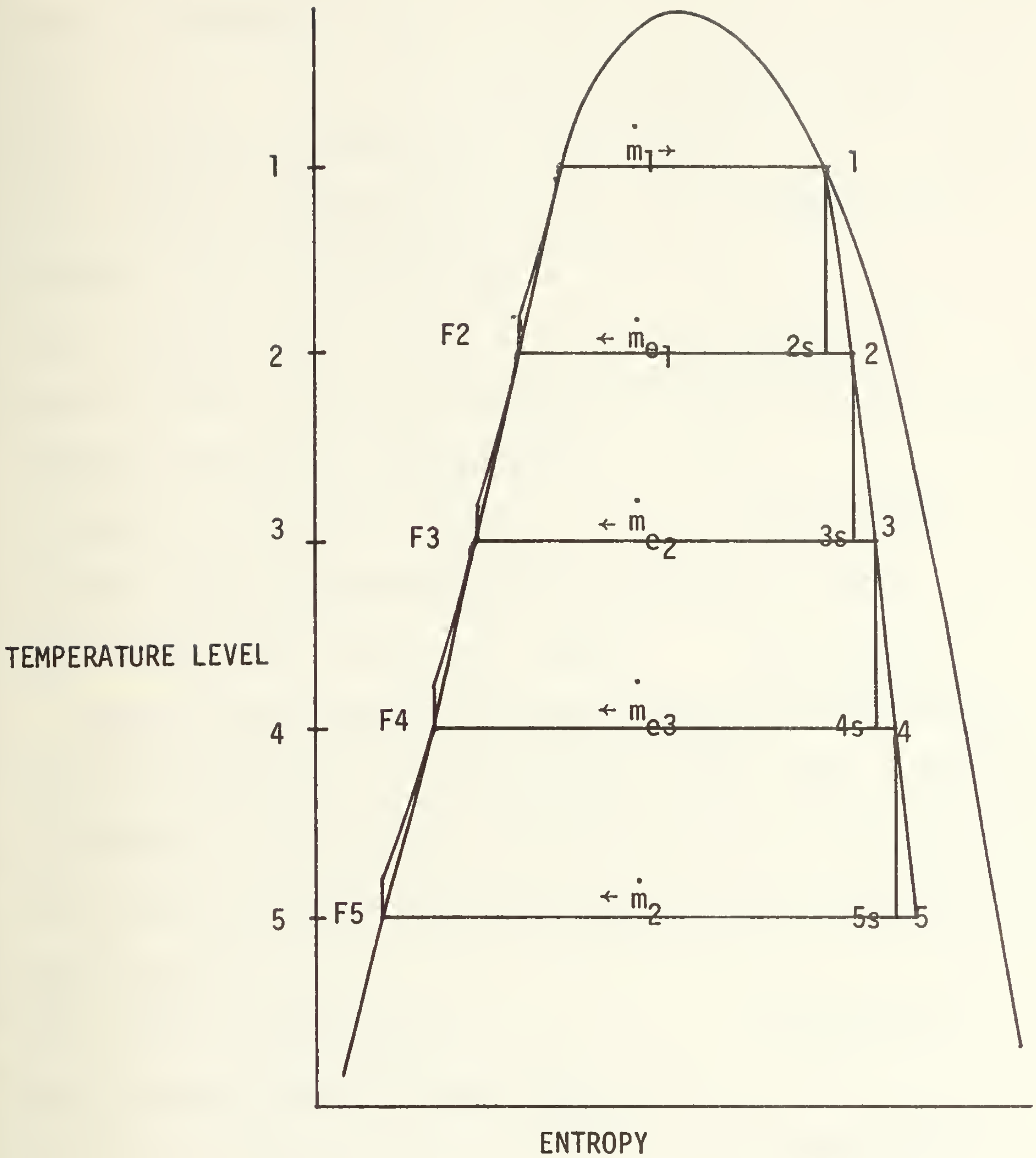


FIGURE 16 TEMPERATURE-ENTROPY DIAGRAM WITH ANALYSIS NOMENCLATURE FOR NH_3 CYCLE

The ideal enthalpy drop is found by considering the expansion in the turbine stage to be isentropic. Therefore, using the first stage as an example:

$s(2s) = s(1)$ (equation 55)

and the quality at state 2s is:

$x(2s) = (s(2s) - s_f(2))/s_{fg}(2)$ (equation 56)

therefore, the enthalpy of state 2s is:

$h(2s) = h_f(2) + h_{fg}(2) \cdot x(2s)$ (equation 57)

However, since in reality the expansion is not isentropic, η_{is}, the isentropic efficiency is introduced:

$\eta_{is} = (h(1) - h(2))/(h(1) - h(2s))$ (equation 58)

therefore the actual enthalpy drop in the first stage is:

$\Delta h_{12} = h(1) - h(2) = \eta_{is}\ (h(1) - h(2s))$ (equation 59)

The loss due to wetness corresponds to one percent power loss for each one percent of average moisture in that stage. Conversely, the wetness losses could be accounted for by multiplying the enthalpy drop by the average quality in the stage. For the first stage,the exit quality x(2) is:

$x(2) = (h(2) - h_f(2))/\ h_{fg}(2)$ (equation 60)

and the average quality in the first stage

$\bar{x}_1 = (x(1) + x(2))/2$ (equation 61)

The mechanical, stage, and exit losses are each assumed to be one percent. Expressing these three factors in one expression called mechanical efficiency, η_m which is simply one minus the three one percent loss, thus $\eta_m = .97$. (equation 62)

Therefore, the power extracted from the first stage of the ammonia turbine P_{T_1}, is:

$$P_{T_1} = \dot{m}_1(h(1) - h(2)) \cdot \eta_m \cdot (x(1) + x(2))/2) \qquad \text{(equation 63)}$$

The same analysis may be applied to the final three stages of the ammonia turbine, provided the proper mass flow rate for each stage is used. Recall that each successive stage has a smaller flow rate due to feedheating extractions.

The purpose of feedheating is to improve the cycle efficiency by increasing the average temperature of heat addition to the cycle. This is accomplished by extracting some of the working fluid and utilizing it to heat the boiler feed. The type of feedheater used in the analysis is an open-type, in which the feed is mixed with the extracted fluid.

The extracted fluid mixes with the feed after the feed pump exit. The mass flow rate of the extraction is adjusted so that the mixture is at the saturated liquid stage. For example, the condensate (state F5 of figure 16) is pumped to the pressure corresponding to saturation at temperature level four. Ammonia is extracted at state 4 and mixed with the feed until the mixture is at the state F4. This process is repreated until the feed is at the boiler inlet condition.

An energy balance in each feedheater states:

$$\dot{m}_{feed}\ \Delta h_{feed} = \dot{m}_{extraction}\ \Delta h_{extraction} \qquad \text{(equation 64)}$$

$$\text{or } \dot{m}_{extraction} = \dot{m}_{feed}\ \Delta h_{feed} / \Delta h_{extraction} \qquad \text{(equation 65)}$$

By substituting, using the symbols from figure 16 equation (65) becomes:

$\dot{m}_{e_1} = (1 - \dot{m}_{e_1})\ (h_f(2) - (h_f(3) + W_p(3))/(h(2) - h_f(2))$ (equation 66)

$\dot{m}_{e2} = (1 - \dot{m}_{e1} - \dot{m}_{e_2})(h_f(3) - h_f(4) + W_p(4))/(h(3) - h_f(3))$ (eq. 67)

$\dot{m}_{e3} = (1 - \dot{m}_{e1} - \dot{m}_{e2} - \dot{m}_{e_3})(h_f(4) - (h_f(5) + W_p(5))/(h(4) - h_f(4))$ (eq. 68)

where $\dot{m}_1 = 1$ and $\dot{m}_2 = 1 - \dot{m}_{e_1} - \dot{m}_{e_2} - \dot{m}_{e_3}$.

The term W_p which appears in these three equations is the enthalpy rise associated with the pump work. The subscript number indicates the temperature level at the pump inlet. The pump work, assuming the specific volume change across the pump is negligible is:

$W_p\ (n) = v_f(n)\ (P_{n+1} - P_n)/\eta_p \times 144/778$ BTU/lbm   (equation 69)

where n is the temperature level at the pump inlet, and ηp is the pump isentropic efficiency.

The heat added to the ammonia cycle is the product of the mass flow rate and the enthalpy rise in the ammonia boiler. The heat added

$Q_{A\ NH_3} = \dot{m}_1(h(1) - (h_f(2) + W_p(2))$ (equation 70)

to the ammonia cycle is also the heat rejected by the steam cycle. Combining equations (54) and (70),

$Q_{A\ NH_3} = Q_{A\ steam}(1-\eta_{th\ steam}) = \dot{m}_1(h(1) - (h_f(2) + W_p(2))$.   (eq. 71)

The left hand side of this equation is fixed by the steam condensing temperature T_S. The enthalpies on the right side are functions of the ammonia boiling and condensing temperatures, TF(1) and TF(5) respectively. Therefore, once these three variables are fixed, $\dot{m}_1$ the mass flow rate through the ammonia boiler is also fixed. Recall that $\dot{m}_1$ is an important consideration in the heat transfer analysis of the steam condenser/ammonia boiler.

The thermal efficiency of the ammonia cycle is:

$$\eta_{th\ NH_3} = \frac{\sum_{i=1}^{4} P_{T_i} - \sum_{i=2}^{5} \dot{m}_i W_{P_i}}{Q_{A\ NH_3}} = \frac{\text{Net Power }(NH_3)}{Q_{A\ NH_3}} \quad \text{(equation 72)}$$

c. The Combined Cycle

The thermal efficiency of the combined cycle, $\eta_{th\ cc}$ is the total of the net power outputs for both cycles divided by the heat added to the cycle:

$$\eta_{th\ cc} = \frac{P_{st} + P_{NH_3}}{Q_{A\ steam}} \quad \text{(equation 73)}$$

recall that $P_{st} = Q_{A\ steam} \times \eta_{th\ steam}$ (equation 74)

and $P_{NH_3} = Q_{A\ NH_3} \times \eta_{th\ NH_3}$ (equation 75)

where $Q_{A\ NH_3} = Q_{A\ steam}(1 - \eta_{th\ steam})$ (equation 76)

therefore, $\eta_{th\ cc} = \eta_{steam} + (1 - \eta_{steam})\eta_{th\ NH_3}$ (equation 77)

There are several other factors which cause the overall efficiency to be lower than the thermal efficiency. The first is the pumping requirements for the ammonia condenser discussed in the preceding chapter, and labelled PP. The second consideration is the power loss associated with converting the mechanical power of the turbines to electrical power, and may be expressed as η_g, the generator efficiency. The third factor is the power that must be expended to prepare the coal and to drive the auxiliary equipment in the plant. This factor is expressed as η_a, the auxiliary efficiency.

The fourth and final factor is the boiler efficiency η_b which is a measure of how much of the energy released in the combustion of the fuel actually is transferred to the working fluid.

The cycle efficiency η_c, is the efficiency of the cycle including the condenser pumping power:

$$\eta_c = \frac{(P_{st} + P_{NH_3}) - PP}{Q_{A\ steam}} = \eta_{th\ cc} - PP/\ Q_{A\ steam} \qquad \text{(equation 78)}$$

Thus the overall plant efficiency is:

$$\eta_{OA} = \eta_c \times \eta_g \times \eta_a \times \eta_b \qquad \text{(equation 79)}$$

In this chapter, the thermodynamic analysis of the combined cycle was undertaken. In the next chapter, the thermodynanic and component considerations are combined in the economic analysis.

VI. ECONOMIC ANALYSIS

The economic factors of concern in the analysis are the capital and operating expenses. For simplicity, only the cost differences between the combined cycle plant and 'Bull Run' will be considered.

The significant capital expenses are:

(1) Turbomachinery - specifically the ammonia and LP steam turbines.

(2) Heat exchangers - specifically the steam condenser/ammonia boiler.

(3) Financing costs

The significant operating expense is the fuel cost which is derived from the cycle overall efficiency.

The major difference between 'Bull Run' and the combined cycle in turbomachinery costs is the addition of the ammonia turbine and the removal of the last stage or two of the low pressure steam turbine.

Due to the large specific volume of steam at condensing conditions in conventional power plants, the last few stages of the LP steam turbine are quite large and costly. Data from DeLaval[18] indicates that the cost of a steam turbine varies approximately linearly with power output as the backpressure is increased. This data fixes the cost variation with power at \$31 per kilowatt of mechanical power output.

The vapor pressure of ammonia at condensing conditions is much

higher than that of steam. Consequently, the specific volume of ammonia is 100 - 300 times lower than steam, causing the ammonia turbine to be smaller and cheaper than the steam turbine it replaces. The selection of \$5/kw[19] for the ammonia turbine is consistent with previous studies and existing NH_3 turbine costs corrected to 1976 dollars.

The net power output of the combined cycle under consideration is the same as that of the 'Bull Run' plant scaled up to 1000 MW. Therefore, any power lost by the steam turbine by the addition of the bottoming cycle must be compensated for by the ammonia turbine. The difference in turbine costs between the binary-cycle plant and 'Bull Run' may be expressed as the product of the NH_3 turbine power and the difference in cost per kilowatt between the steam and ammonia turbines. Thus,

TURBOMACHINERY COST SAVINGS = P_{NH_3} x (\$31/kw - \$5/kw), (equation 80)

where P_{NH_3} was derived as a function of the operating parameters in the last chapter.

The most significant heat exchanger cost in the combined cycle is that of the steam condenser/ammonia boiler. The total area of the SC/AB is the sum of the boiling region area, A_B, and the non-boiling area A_{NB}, introduced in chapter four.

$$A_T = A_B + A_{NB} \quad \text{(equation 81)}$$

Utilizing heat exchanger cost information[20], the cost of the SC/AB

per unit area is assumed to be $\$8/ft^2$. The cost of the steam condenser/ammonia boiler is:

$$\text{COST SC/AB} = A_T \times \$8/ft^2 \qquad \text{(equation 82)}$$

Combining equations (80) and (82) the additional initial capital expense of the combined cycle is:

$$\text{INITIAL CAPITAL COST} = A_T \times \$8/ft^2 - P_{NH_3} \times \$26/kw \qquad \text{(equation 83)}$$

In practice, the funds for capital equipment are borrowed, incurring finance charges.

For a loan where the annual repayment rate is the same for every year, the total cost of capital equipment including finance charges is:

$$\begin{matrix}\text{TOTAL COST} \\ \text{OF EQUIPMENT}\end{matrix} = \begin{matrix}\text{INITIAL CAPITAL} \\ \text{COST}\end{matrix} \times \frac{i\ (1 + i)^n}{(1 + i)^n - 1} \times n \qquad \text{(equation 84)}$$

where n is the loan duration and i is the interest rate per annum.

It must be noted that the turbomachinery cost per kw and heat exchanger cost per square foot of surface area represent 1976 values. The analysis and optimization procedure is easily modified to reflect price changes.

The operating costs for the combined cycle when compared with those of the unmodified 'Bull Run' plant are reflected in the differences in fuel consumption between the two cases. The heat input to a cycle is the product of the fuel mass flow rate and the heating value of the fuel: $\text{HEAT INPUT} = \dot{m}_{coal} \times HHV_{coal}$ (equation 85)

The power output is the product of the heat input and the overall efficiency, $\text{POWER} = \text{HEAT INPUT}/\eta_{OA}$ (equation 86)

Combining equations (85) and (86), the coal mass flow rate is:

$$\dot{m}_{coal} = \frac{\text{POWER OUTPUT}}{\eta_{OA} \times HHV_{coal}} \qquad \text{(equation 87)}$$

The difference in coal flow between the 'Bull Run' plant and the combined cycle is:

$$\dot{m}_{BULL\ RUN} - \dot{m}_{cc} = \frac{(\text{POWER OUTPUT})_{BR}}{\eta_{OA_{BULL\ RUN}} \times HHV_{coal}} - \frac{(\text{POWER OUTPUT})_{cc}}{\eta_{OA_{cc}} \times HHV_{coal}} \qquad \text{(equation 88)}$$

Since the analysis has assumed that the steam portion of the combined cycle except for the condensing region is identical to 'Bull Run' and that both plants have the same net power, equation (88) may be rewritten as follows:

$$\dot{m}_{BR} - \dot{m}_{cc} = \frac{\text{POWER OUTPUT}}{HHV_{coal} \times \eta_a \times \eta_g \times \eta_b} \left[\frac{1}{(\eta_c)_{BR}} - \frac{1}{(\eta_c)_{cc}} \right] \qquad \text{(equation 89)}$$

where η_a, η_g, and η_b refer to equation (79) and η_c is the cycle efficiency, a modification of thermal efficiency which includes condenser pumping requirements.

Equation (89) indicates that if the combined cycle efficiency $(\eta_c)_{cc}$ is higher than the 'Bull Run' cycle efficiency $(\eta_c)_{BR}$ fuel will be saved.

The economic gain of reduced fuel consumption is:

$$\text{DOLLAR SAVINGS PER HOUR} = \text{COAL COST \$/lbm} \times (\dot{m}_{BR} - \dot{m}_{cc}) \frac{\text{lbm}}{\text{hr}} \qquad \text{(eq. 90)}$$

Equation (90) could be used to extrapolate the savings over the plant lifetime, however the term 'power output' appears in equation (89). It would be too optimistic to design a combined cycle based on 100%

load demand over the entire plant life. The quantity load factor, LF, expresses the ratio of the expected average power output of the cycle to its maximum capacity. Equation (89) becomes:

$$\dot{m}'_{BR} - \dot{m}'_{cc} = \frac{\text{MAXIMUM POWER OUTPUT x LF}}{HHV_{coal} \times \eta_a \times \eta_g \times \eta_b} \left[\frac{1}{\eta c_{BR}} - \frac{1}{\eta c_{cc}} \right] \quad \text{(eq. 91)}$$

Then the fuel savings in dollars over the plant lifetime is:

$$\text{LIFETIME FUEL SAVINGS} = \text{COAL COST \$/Ton} \times (\dot{m}'_{BR} - \dot{m}'_{cc}) \frac{\text{lbm}}{\text{hr}} \times \frac{1 \text{ Ton}}{2000 \text{ lbm}} \times 8760 \frac{\text{hr}}{\text{yr}} \times n \text{ yrs} \quad \text{(eq. 92)}$$

where n is the expected plant lifetime in years, as well as the loan duration.

Combining the operating and capital expenses into one equation, the life cycle benefit in dollars of the combined cycle over the unmodified 'Bull Run' plant is:

$$\text{NET SAVINGS} = \text{LIFETIME FUEL SAVINGS} - \text{TOTAL COST OF EQUIPMENT (FROM EQUATION 84)} \quad \text{(eq. 93)}$$

The purpose of the optimization procedure is to maximize the 'NET SAVINGS' of equation (93), and determine the operating parameters, in terms of the three independent variables T_S, TF(1) and TF(5), which yield the maximum economic benefit.

VII. OPTIMIZATION PROCEDURE

In this chapter, the procedure for the selection of the optimum design bottoming cycle based on economic and engineering factors will be outlined.

The optimum design may be specified in terms of three operating parameters previously identified as the independent variables in the analysis, they are:

(1) The steam condensing temperature, T_S.

(2) The ammonia boiling temperature, TF(1).

(3) The ammonia condensing temperature, TF(5).

Once values for these variables are chosen, a thermodynamic analysis of the combined cycle is undertaken which yields the combined cycle thermal efficiency and the power split as the output. Power split is defined as the amount of power produced by each of the two cycles.

Next, the heat transfer analysis for the steam condenser/ammonia boiler specifies the size of the heat exchanger. The ammonia condenser is then analyzed with the condenser pumping power as the result.

The information derived from these analyses is combined in equation (93), which combines capital and operating costs into a value of net savings over the plant lifetime.

A flow diagram of the optimization procedure is shown as figure 17. Note that the value of net savings resulting from the analysis is

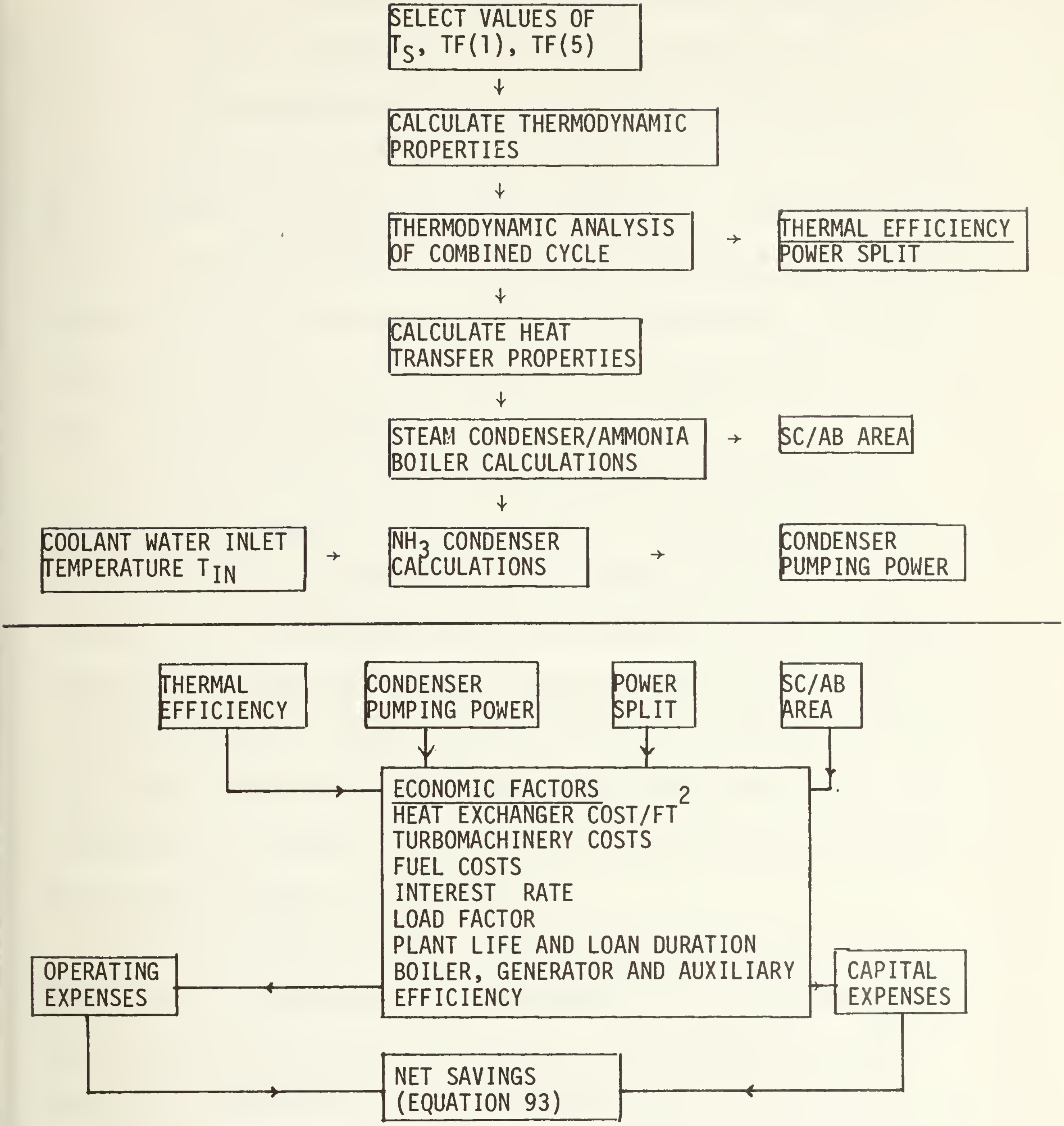


FIGURE 17 OPTIMIZATION PROCEDURE FLOW CHART

very sensitive to the economic factors used in the cost computations. The most significant factors are fuel and heat exchanger costs.

Thus far, the optimization has produced a value of net savings of the combined cycle plant over that of the 'Bull Run' plant scaled up to 1000 MW for one set of the three variables T_S, TF(1), and TF(5). In order to determine which set will yield the optimum plant, the entire analysis must be carried out for all possible combinations of the three independent parameters within the range of values where the bottoming cycle is expected to operate.

The analysis may be accomplished independent of environmental conditions with regard to selection of values of the steam condensing temperature, T_S, and the ammonia boiling temperature TF(1). However, the selection of the ammonia condensing temperature, TF(5) is very much affected by the temperature of the condenser coolant water at the inlet.

As discussed in chapter four, the relationship between the condenser water inlet temperature and the ammonia condensing temperature is restricted by considerations of pumping power and tube corrosion resulting from high water mass flow rates and velocities.

Utilizing equations (13) and (78) the cycle efficiency may be calculated for various values of temperature difference between the condensing ammonia and the inlet water for fixed parameters T_S, TF(1), and TF(5). Additionally, the water velocity through the tubes as a function of the same parameters may be calculated. The results of these calculations are plotted as figure 18. The values of steam condensing temperature, ammonia boiling temperatures, and ammonia condensing tem-

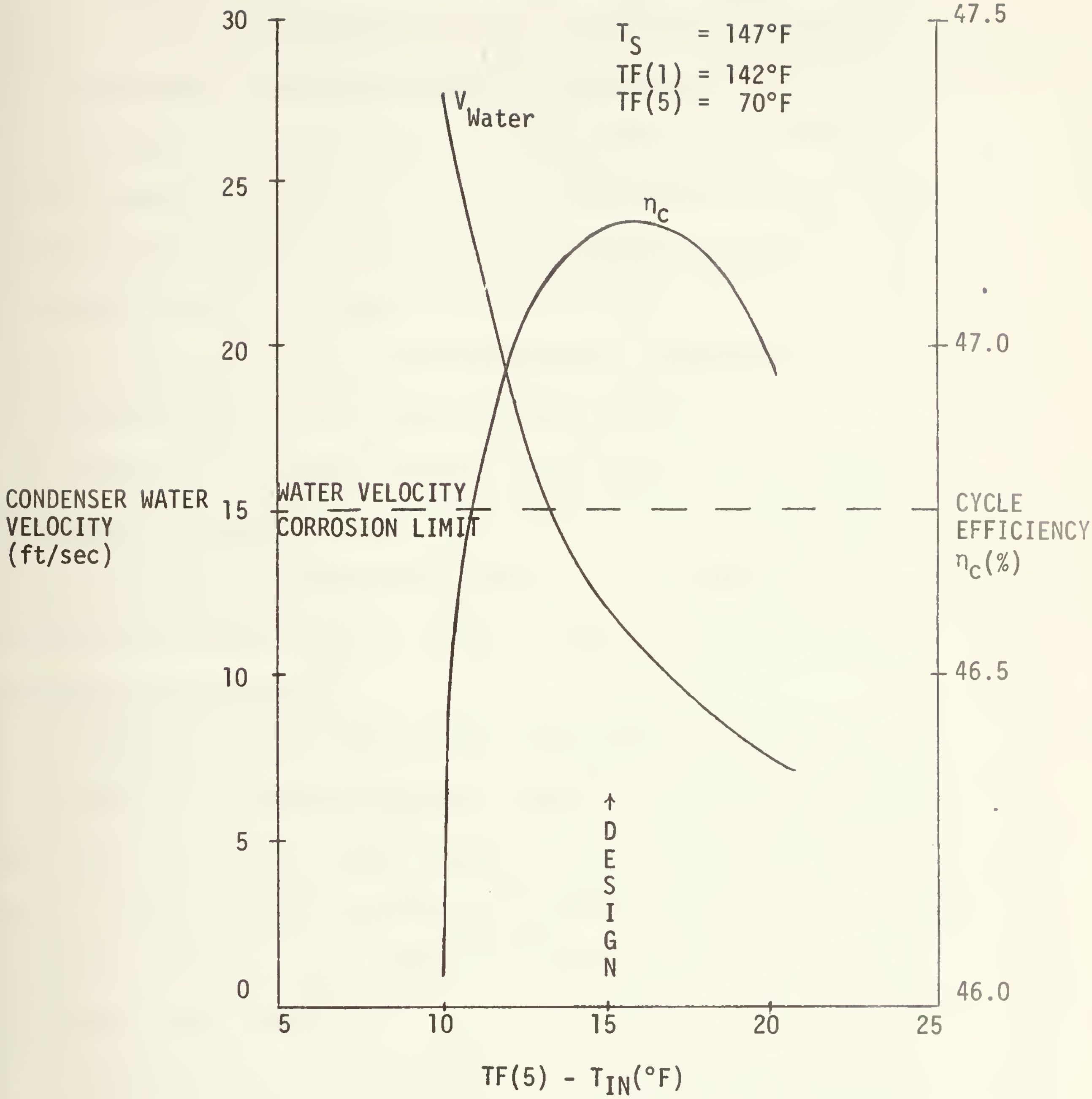


FIGURE 18 CONDENSER WATER VELOCITY AND CYCLE EFFICIENCY VS. DIFFERENCE BETWEEN NH_3 CONDENSING TEMPERATURE AND WATER INLET TEMPERATURE

perature used for figure 18 are representative, near optimum values. Notice that the cycle efficiency is at a maximum at approximately a 15°F temperature difference between the condensing ammonia and the inlet coolant water. Furthermore, the same temperature difference yields a water velocity of 12.5 ft/sec, close to the corrosion limit of 15.0 ft/sec. Thus it appears that the optimum selection of TF(5), the ammonia condensing temperature is 15°F above the available water source temperature derived from environmental information. This conclusion agrees with earlier condenser design studies.[21]

Replacing the ammonia condensing temperature TF(5) as one of the independent parameters by the water inlet temperature T_{IN} and keeping the 15°F difference between TF(5) and T_{IN} for all possible designs, the optimization computations for different sets of the independent parameters may now proceed.

Since the combined cycle under consideration is compared to the 'Bull Run' plant, the environmental conditions of the 'Bull Run' plant are used. At the 'Bull Run' site, the average condenser inlet temperature is 55°F[22], and this value is incorporated into the optimum design analysis. However, other values of condenser inlet temperature are included in the results.

Therefore, only the steam condensing and ammonia boiling temperatures need be varied to find the optimum design. In practice, the steam condensing temperature is fixed and the temperature difference between the condensing steam and boiling ammonia is varied. To present the results on a non-dimensional basis, this temperature difference is

expressed as the difference between the steam condensing temperature and the entropy averaged temperature of heat addition to the ammonia, $\overline{T}$, divided by the steam condensing temperature,

$$\Delta\overline{T}/T_S = (T_S - \overline{T})/\ T_S \qquad \text{(equation 94)}$$

where $\overline{T}$ is a function of the ammonia boiler inlet and outlet conditions.

In order to carry out the optimization procedure, the many engineering and economic factors previously discussed must be given values. The optimization computer program (see appendix one), will carry out the analysis for any values of these factors. This flexibility enables this optimization to be used for future bottoming cycle design utilizing up-to-date economic considerations. The symbols and values used in this analysis are shown in table 4.

Thus far, the derivation of the analysis and the procedure for identifying the optimum design combined cycle have been presented. In the next chapter, the results of the optimization calculations are presented along with the specifications of the optimum bottoming cycle compatable with the 'Bull Run' plant.

EFFICIENCIES	TEXT SYMBOL	COMPUTER SYMBOL	VALUE
1. Auxiliary Efficiency	η_a	NA	94 %
2. Boiler Efficiency	η_b	R4	89 %
3. Bull Run Cycle Efficiency	$(\eta_c)_{BR}$	R5	45.78%
4. Generator Efficiency	η_g	R7	99 %
5. Pump Efficiency	η_p	NP	90 %
6. Turbine Isentropic Efficiency	η_{is}	NS	84 %
7. Turbine Mechanical Efficiency (Includes exit, stage and mechanical losses)	η_M	NM	97 %
HEAT EXCHANGER SPECIFICATIONS			
8. Ammonia Boiler Tube Diameter	D	D2	1"
9. Ammonia Boiler Tubes in Vertical Bank	n	NOT	70
10. Ammonia Condenser Tube Diameter	D	D3	.875"
11. Ammonia Condenser Tubes in Vertical Bank	n	INCLUDED IN A CONSTANT	200
ECONOMIC FACTORS			
12. Coal Cost (Ill. #6)	-	R6	$50/Ton
13. Heat Exchanger Cost	-	R1	$ 8/ft^2
14. Higher Heating Value (Ill. #6)	HHV	RF	10788. BTU/lbm
15. Interest Rate	i	IR	12%
16. Load Factor	LF	R8	.8
17. Loan Duration and Plant Life	n	R9	30 years
18. Turbine Cost (Ammonia)	-	R3	$ 5/kw
19. Turbine Cost (LP Steam)	-	R2	$31/kw

TABLE 4 SUMMARY OF ENGINEERING AND ECONOMIC FACTORS

VIII. OPTIMIZATION RESULTS

Utilizing the optimization procedure outlined in the last chapter, the thermal efficiency and lifetime net savings for each set of combined cycle parameters are calculated. Assuming the average condenser water inlet temperature to be 55°F, and incorporating the engineering and economic factors enumerated in table 4, an optimum design may be predicted.

Figure 19 is a plot of the combined cycle efficiency versus the non-dimensional temperature difference $\Delta\overline{T}/T_S$ for several different values of T_S, the steam condensing temperature. As expected, the cycle efficiency increases as the temperature difference decreases. Depending on the value of $\Delta\overline{T}/T_S$, either a steam condensing temperature of 131°F or 147°F yields the maximum efficiency. This efficiency peak is caused by wetness considerations discussed in chapter 4.

However, as the non-dimensional temperature difference decreases, the area of the steam condenser/ammonia boiler increases. The opposing trends of efficiency and heat exchanger size variation with the non-dimensional temperature difference are the most significant factors in the optimization routine. Figure 20 shows the steam condenser/ammonia boiler areas versus the non-dimensional temperature difference for several values of T_S, the steam condensing temperature. Note that the heat exchanger area is slightly smaller for T_S = 147°F than for 131°F. This trend is caused by the change of the steam properties which

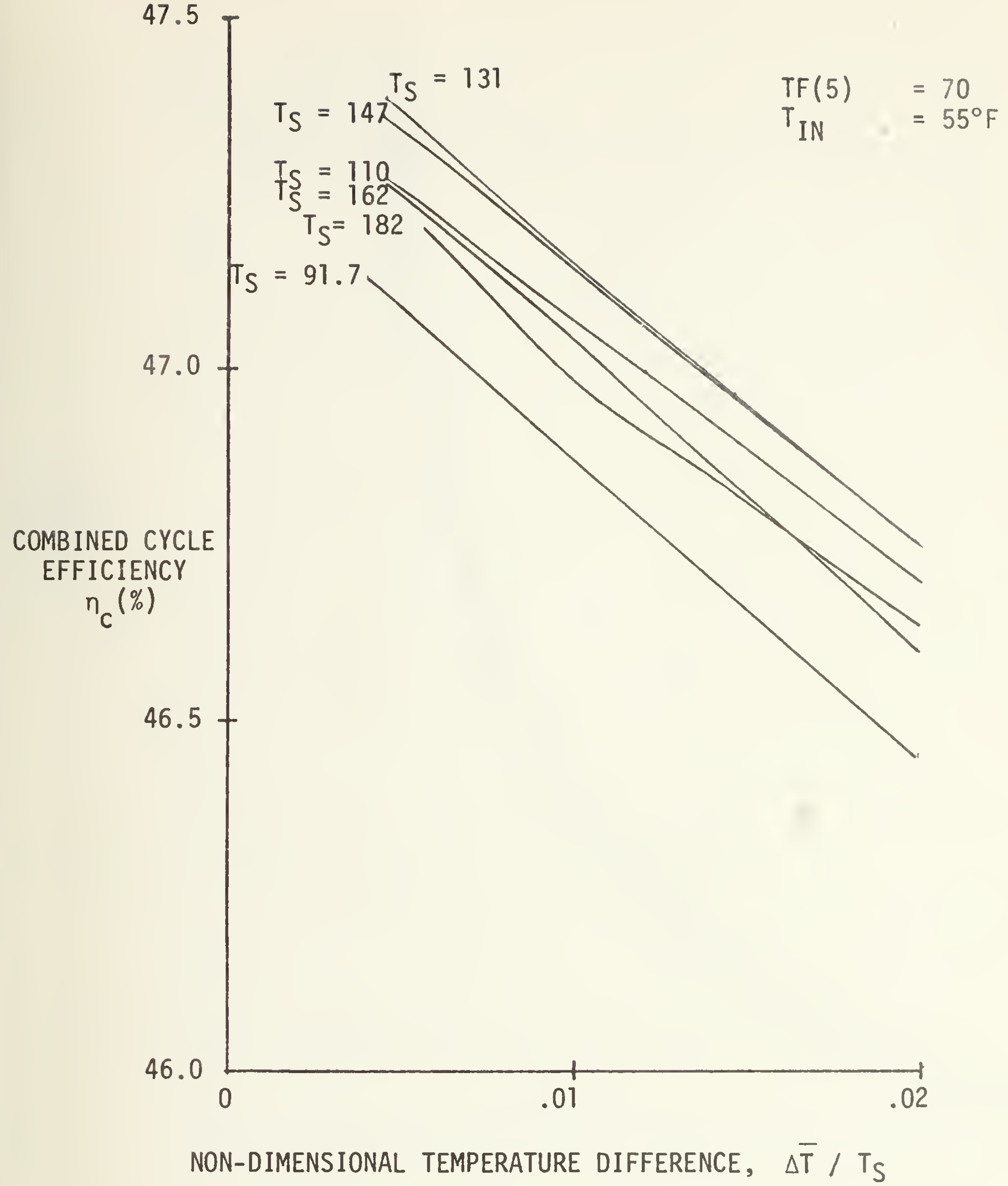


FIGURE 19 COMBINED CYCLE EFFICIENCY VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE FOR VARIOUS VALUES OF STEAM CONDENSING TEMPERATURE. WATER INLET TEMPERATURE = 55°F.

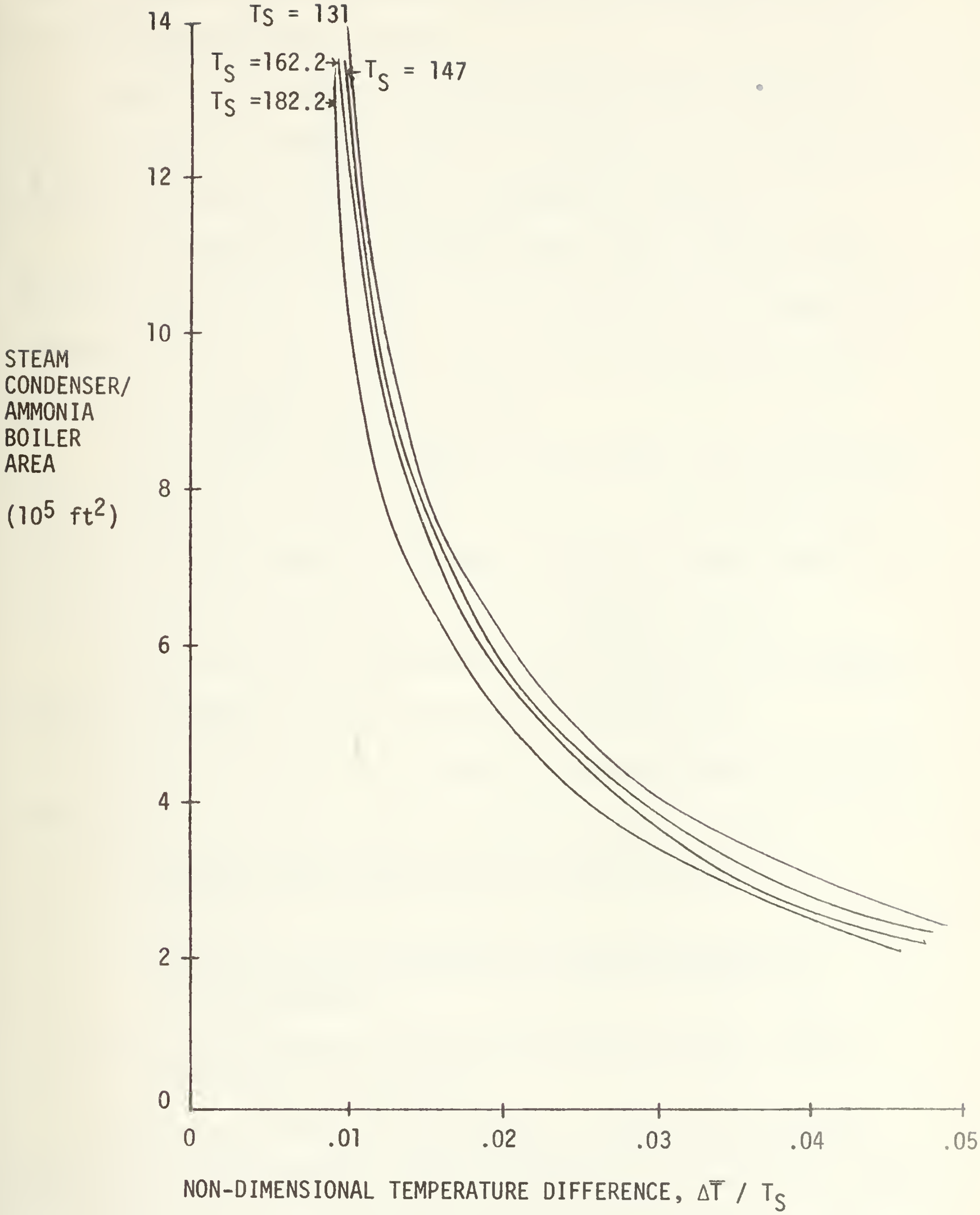


FIGURE 20 STEAM CONDENSER/ AMMONIA BOILER HEAT TRANSFER AREA VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE FOR VARIOUS VALUES OF STEAM CONDENSING TEMPERATURE.

affect the steam side heat transfer coefficient (see equation (23)) as the condensing temperature varies.

Combining the effects of cycle efficiency, heat exchanger size, and the other economic factors in the cost equation (equation (93)), a plot of lifetime cycle savings is presented as figure 21.

It is obvious that the maximum savings occurs where the steam condensing temperature T_S = 147°F, and the non-dimensional temperature difference $\overline{\Delta T} / T_S$ = .09. This value of $\overline{\Delta T} / T_S$ corresponds to an ammonia boiling temperature TF(1) = 142°F.

The predicted lifetime combined cycle savings over the 'Bull Run' plant is $98 million or about .047 cents/kw - hr. The combined cycle efficiency is 47.2%, about 1.4% higher than the 'Bull Run' plant with an initial additional capital cost of only $9.1 million.

The flow chart and thermodynamic specifications of the combined cycle are presented as figure 22 and table 5 respectively.

In order to determine the combined cycle performance in the off-design condition, several assumptions must be made. They are:

(1) Maintain steam condensing and ammonia boiling temperature at design points.

(2) Maintain a 15°F temperature difference between the ammonia condensing temperature and the coolant water inlet temperature.

(3) Utilize DeLaval estimate of off-design ammonia turbine performance (figure 23).

Carrying out the off-design analysis, incorporating the exit loss criteria from figure 23, a plot of combined cycle efficiency versus coolant water inlet temperature is shown as figure 24. For water inlet

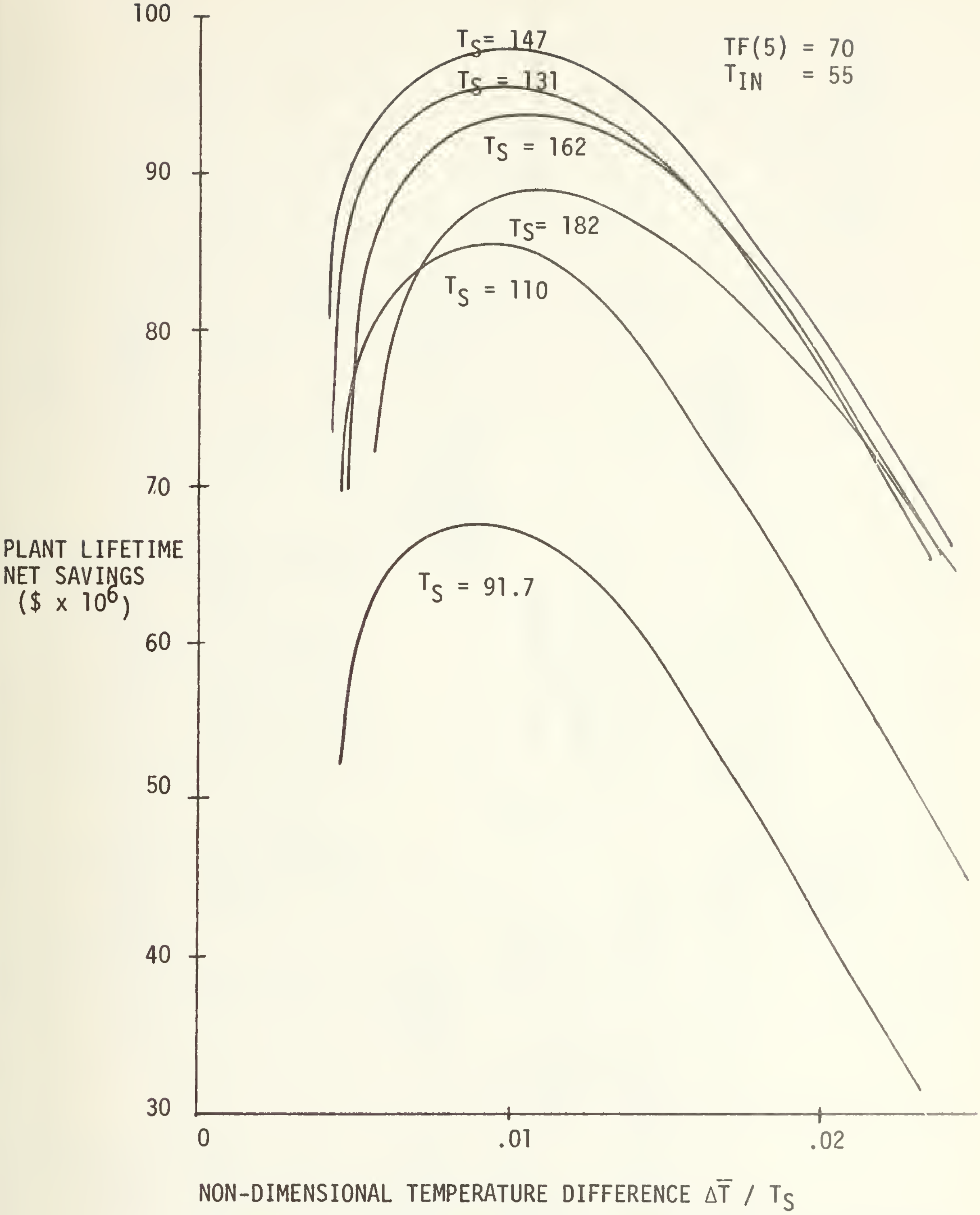


FIGURE 21 PLANT LIFETIME NET SAVINGS VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE FOR VARIOUS VALUES OF STEAM CONDENSING TEMPERATURE WATER INLET TEMPERATURE = 55°F

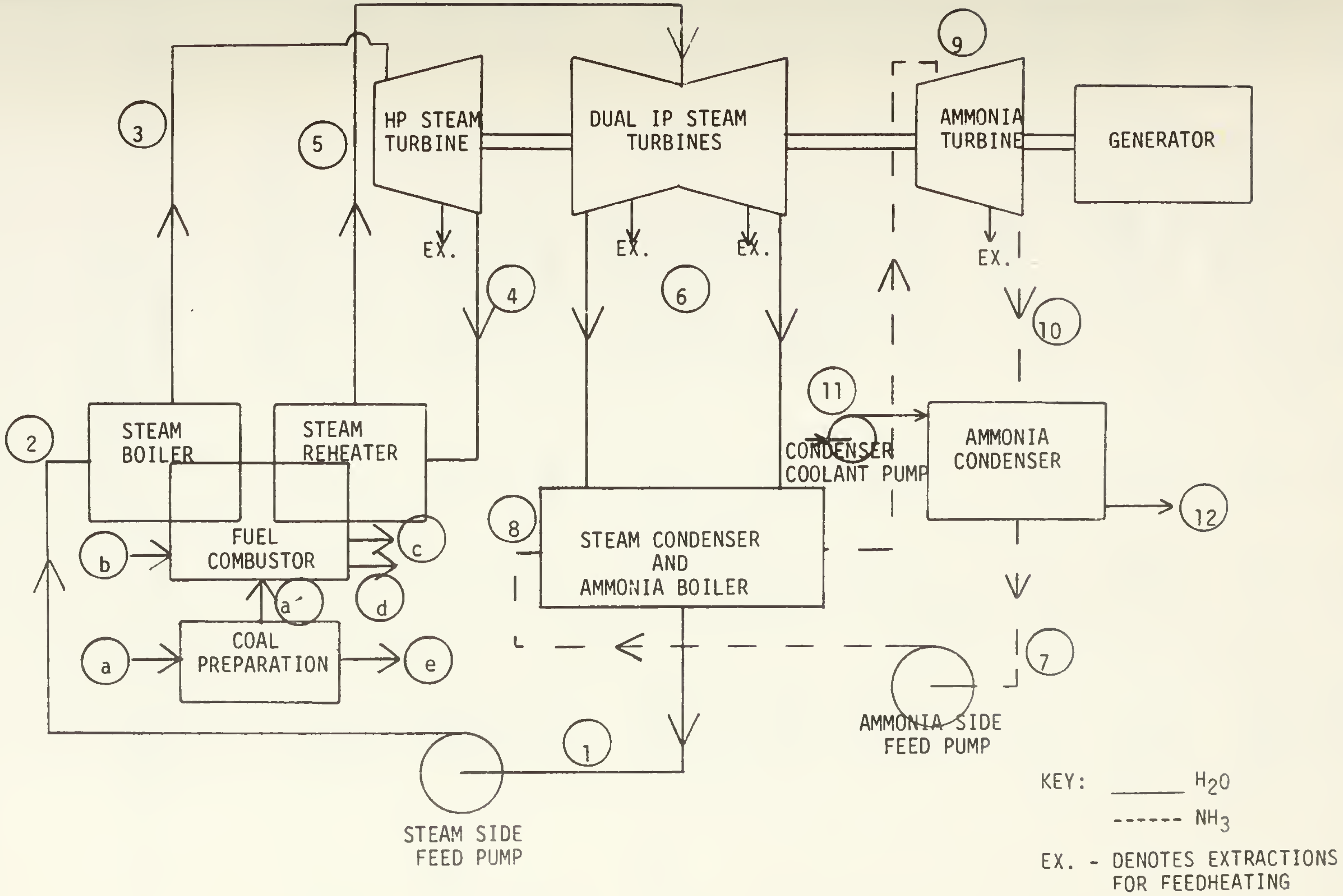


FIGURE 22 COMPONENT FLOW CHART FOR COMBINED CYCLE

STREAM	NO.	DESCRIPTION	P(psi)	T(°F)	h(BTU/lbm)	$\dot{m}(10^6$ lbm/hr)
Fuel	a	Raw fuel (13% H_2O)	14.7	60	HHV 10788	.81
Fuel	a´	Prepared fuel (6% H_2O)	14.7	60	HHV 10788	.75
Air	b	Combustor inlet air	14.7	60	124.3	8.21
Exhaust	c	Products of combustion	14.7	300	189.1	8.82
Ash	d	Ash	14.7	-	-	.08
H_2O	e	Water extracted from fuel	14.7	60	28.0	.06
H_2O	1	Condenser exit	3.45	147	115.0	4.68
H_2O	2*	Boiler inlet	4210	550.1	544.9	7.31
H_2O	3	Boiler exit/HP turbine inlet	3515	1000	1420.8	7.31
H_2O	4	HP turbine exit/ reheater inlet	600	551.7	1255.4	5.58
H_2O	5	Reheater exit/IP turbine inlet	540	1000	1519.6	5.17
H_2O	6	IP turbine exit/ condenser inlet	3.45	147	1078.7	4.68 x = .954
NH_3	7	NH_3 condenser exit	128.8	70	121.1	8.35
NH_3	8*	NH_3 boiler inlet	390.3	124	182.4	10.0
NH_3	9	NH_3 boiler exit	390.3	142	632.4	10.0
NH_3	10	NH_3 turbine exit	128.8	70	612.2	8.35 x = .967
H_2O	11	Coolant water inlet	14.7	55	23.0	513
H_2O	12	Coolant water exit	14.7	63	31.0	513

* INCLUDES FEEDHEATING

TABLE 5(PART 1) THERMODYNAMIC SPECIFICATIONS OF THE COMBINED CYCLE

(PART 2 ON NEXT PAGE)

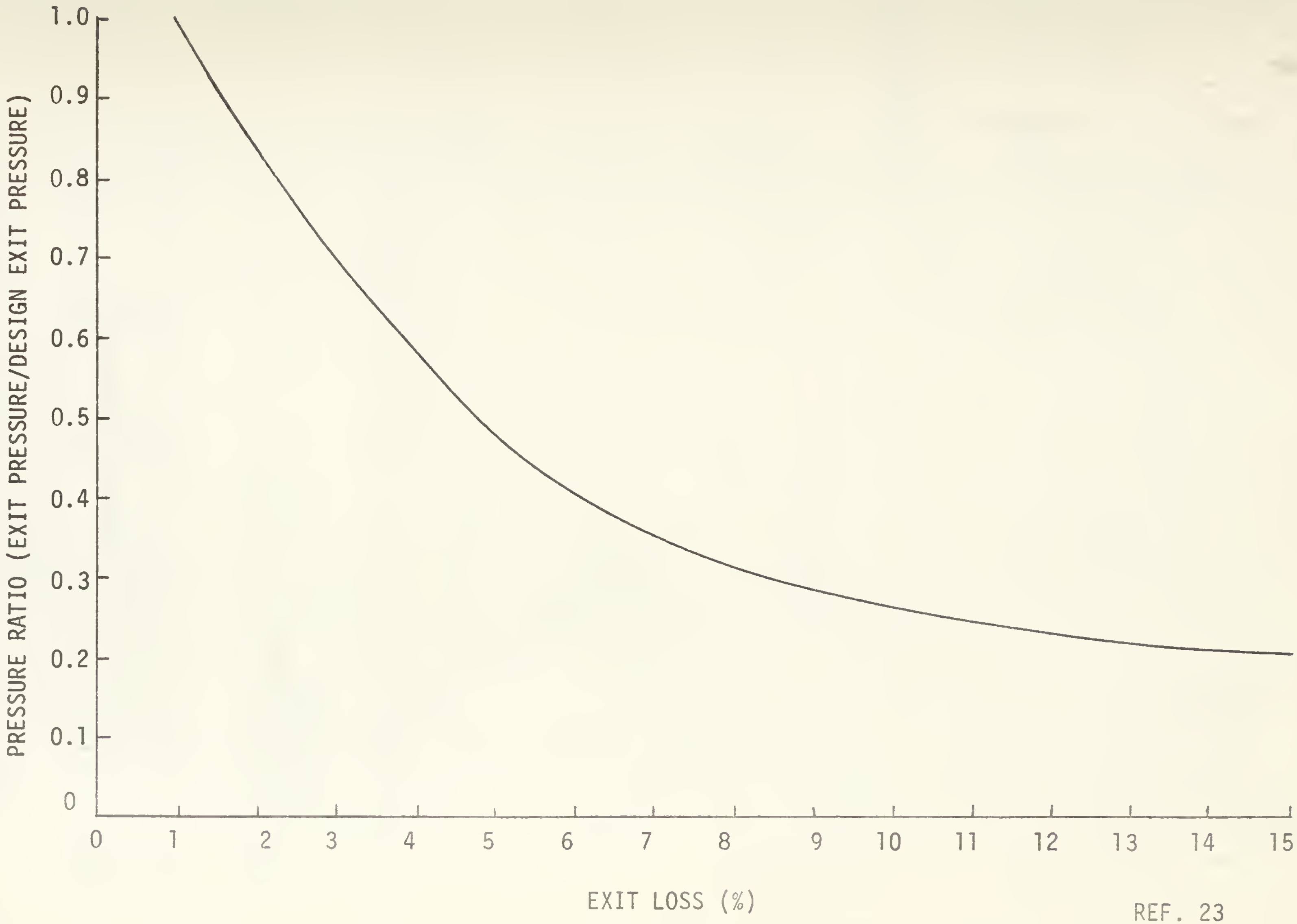
COMPONENT	POWER	HEAT TRANSFER
Steam boiler		1876.0 MW
Steam reheater		400.5 MW
HP steam turbine	272.2 MW	
IP steam turbine	682.6 MW	
Steam condenser/NH_3 boiler		1321.7 MM
Ammonia turbine	123.9 MW	
Ammonia condenser		1201.9 MW

Gross total power	1078.7 MW
Generator losses	-10.8
Auxiliary power	-64.1
Condenser pumps	- 3.8
NET POWER	1000.0 MW

Combined cycle: Thermal efficiency = 47.38%
Cycle efficiency = 47.2%
Overall efficiency = 39.1%

Steam cycle : η_{th} = 41.94%
Ammonia cycle : η_{NH_3} = 9.37%

TABLE 5(PART 2) THERMODYNAMIC SPECIFICATIONS OF THE COMBINED CYCLE



REF. 23

FIGURE 23 EXIT LOSSES AMMONIA TURBINE IN OFF DESIGN CONDITIONS

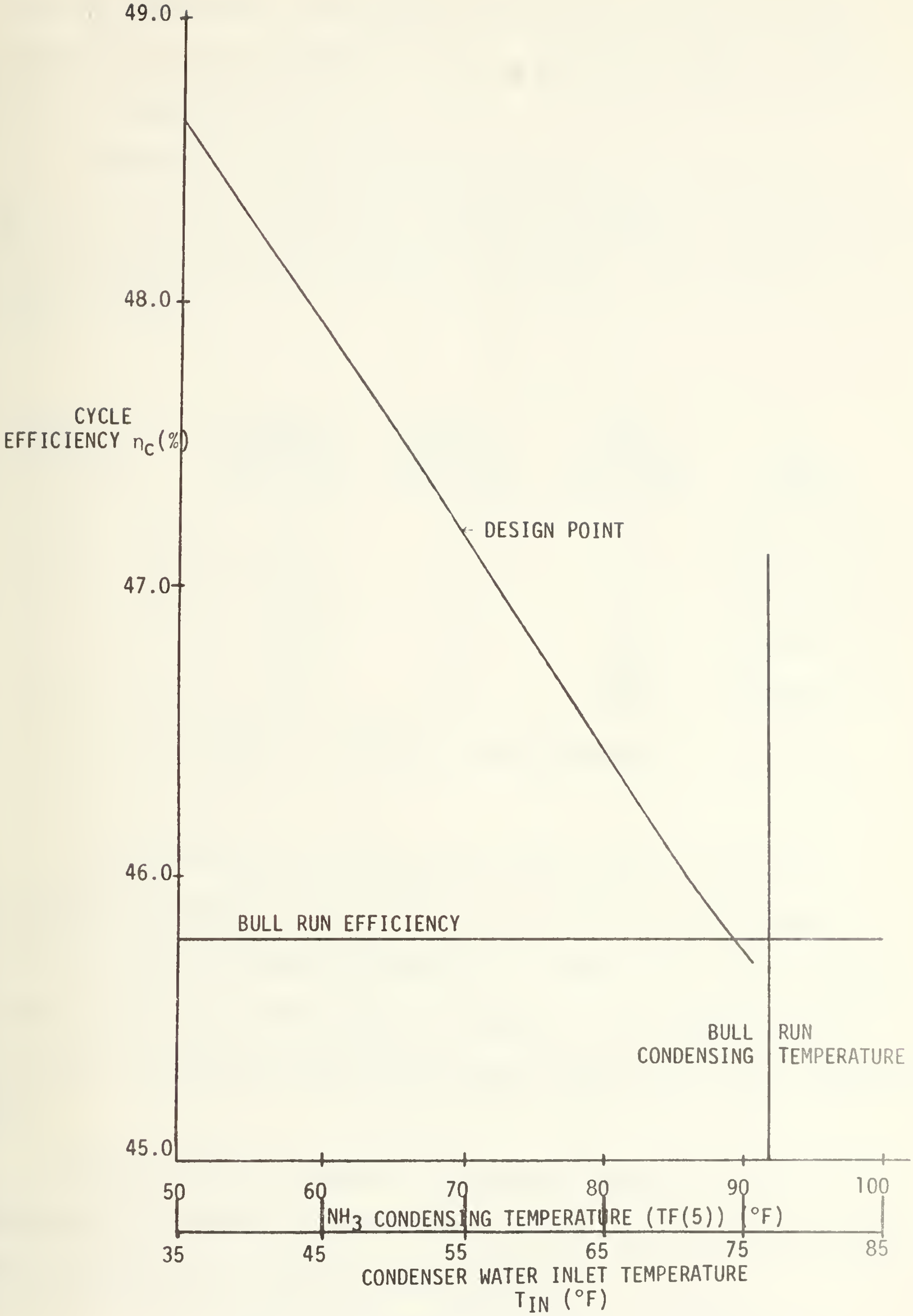


FIGURE 24 OFF-DESIGN PERFORMANCE OF COMBINED CYCLE

temperature above 55°F, the pressure ratio used in figure 23 becomes greater than one. For pressure ratios greater than unity, the exit loss is assumed to be 1%.

Optimization results for condenser water inlet temperatures of 35°F, 45°F, and 65°F are shown in figure 25 through 30. The parameters of the optimum design for each value of condenser inlet temperature are shown in table 6.

CONDENSER INLET TEMPERATURE	STEAM CONDENSING TEMPERATURE	NH_3 CONDENSING TEMPERATURE	NH_3 BOILING TEMPERATURE	NET SAVINGS
35°F	131°F	50°F	126°F	$233 million
45°F	147°F	60°F	142°F	$162 million
55°F	147°F	70°F	142°F	$ 98 million
65°F	147°F	80°F	142°F	$ 29 million

TABLE 6 OPTUMUM DESIGN PARAMETERS

The net savings for each optimum design taken from table 6 is plotted in figure 31. As expected, the combined cycle is most attractive where the ambient temperature is lowest. More significantly, figure 31 indicates at what value of condenser inlet temperature the addition of a bottoming cycle is no longer economically viable, approximately 69°F.

In this chapter, information has been presented to permit the selection of an optimum ammonia bottoming cycle with the available water source temperature as the only input variable.

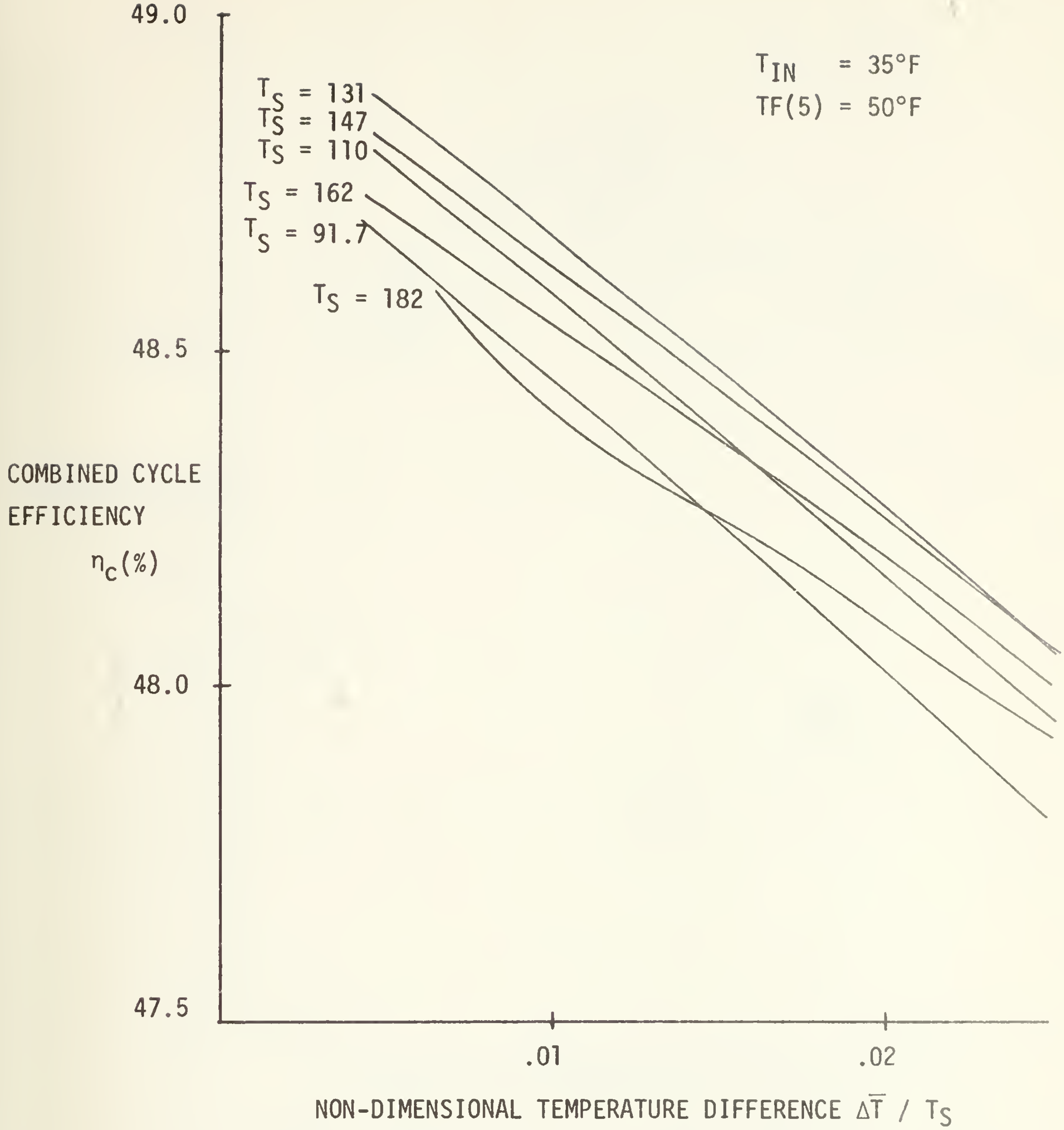


FIGURE 25 COMBINED CYCLE EFFICIENCY VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE
CONDENSER INLET TEMPERATURE = 35°F

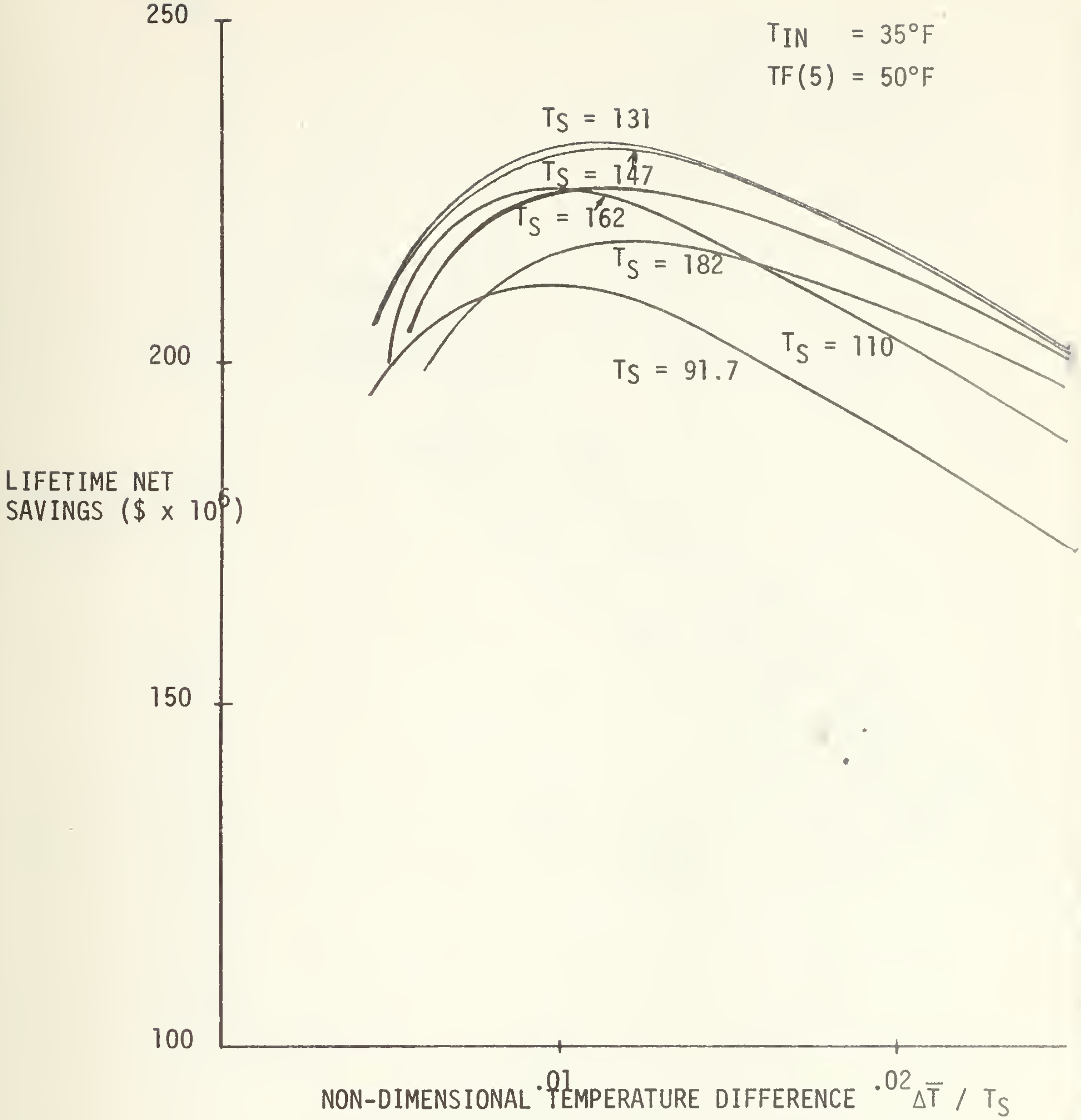


FIGURE 26 COMBINED CYCLE NET SAVINGS VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE
CONDENSER INLET TEMPERATURE = 35°F

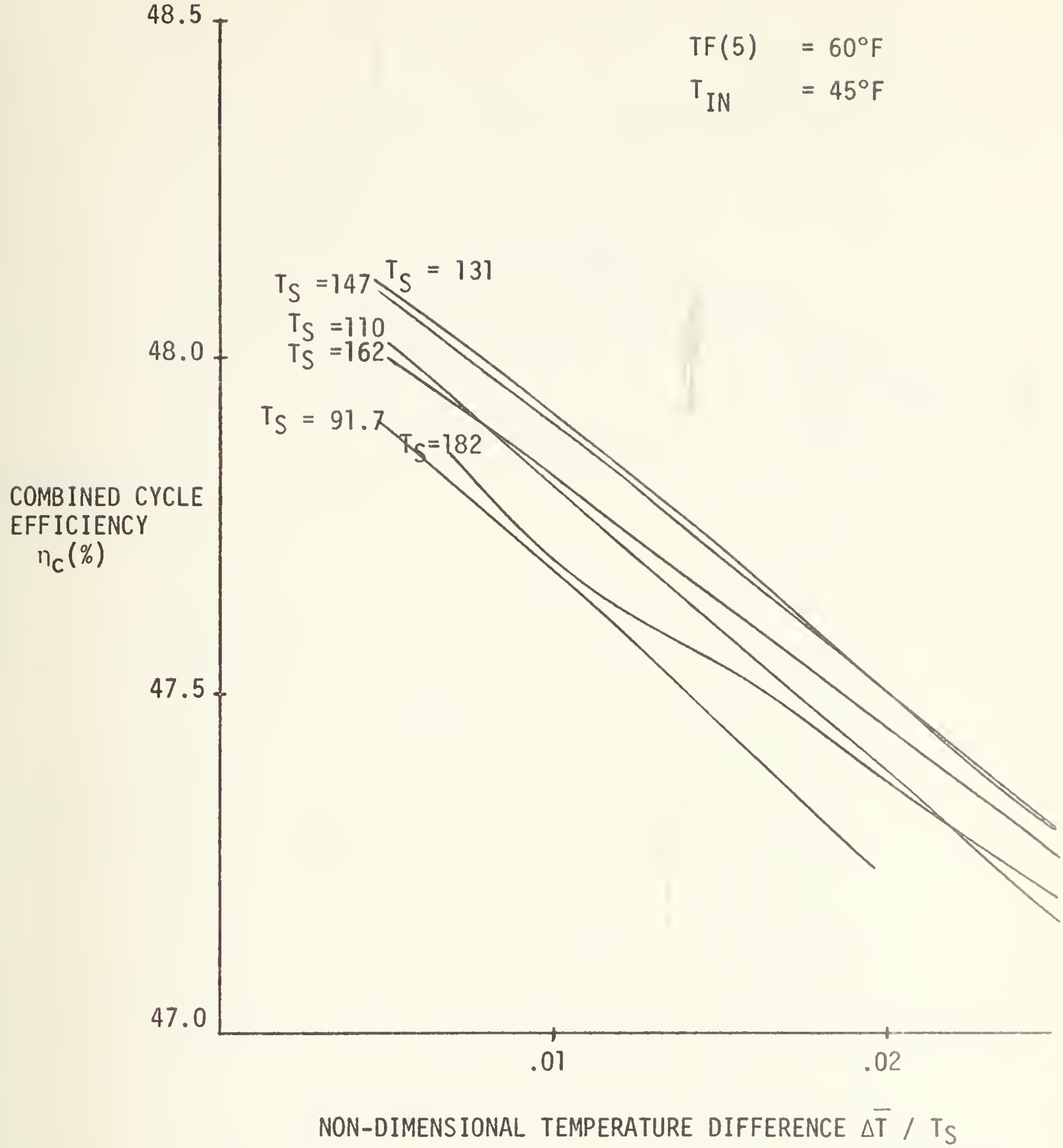


FIGURE 27 COMBINED CYCLE EFFICIENCY VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE
CONDENSER INLET TEMPERATURE = 45°F

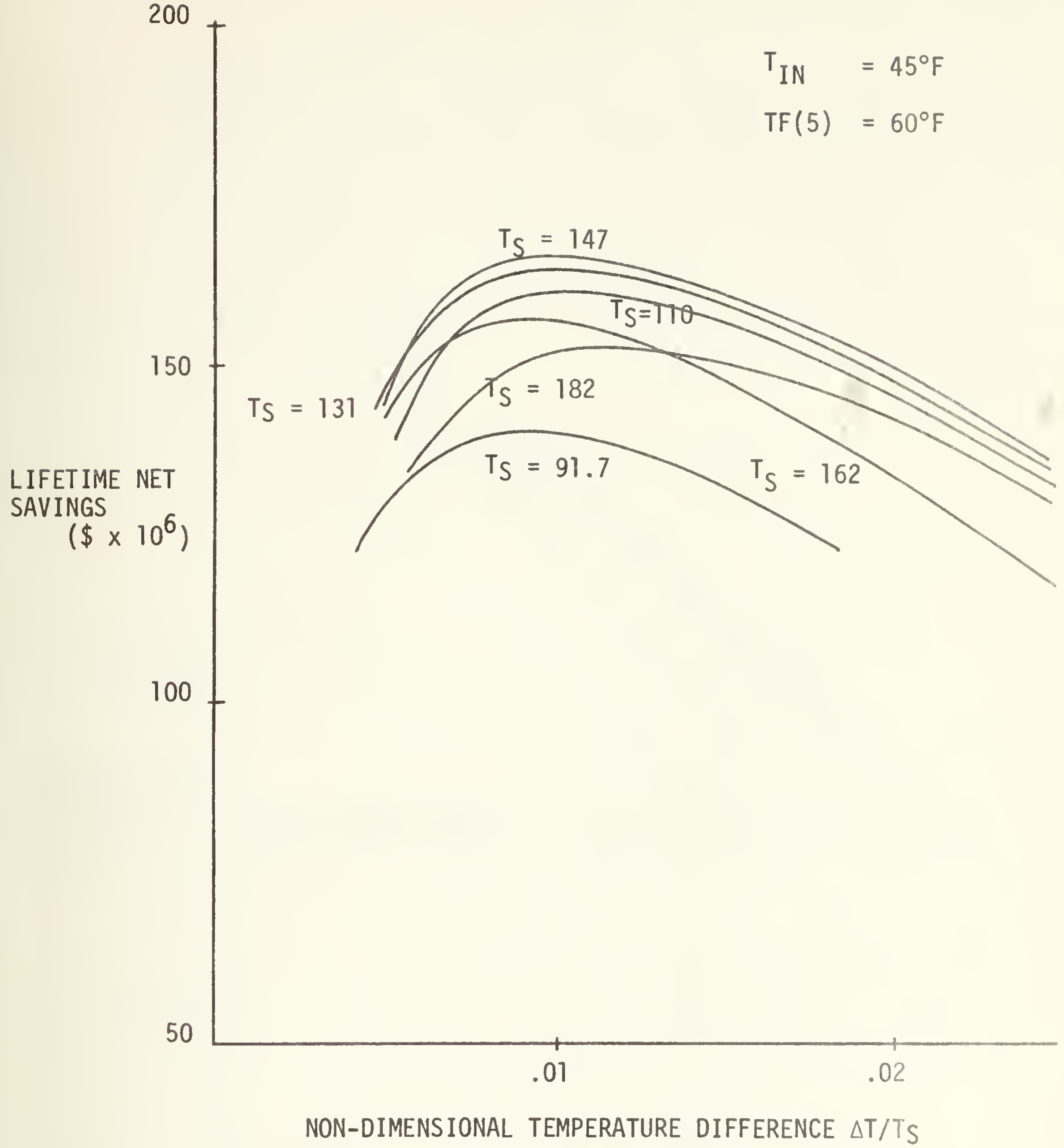


FIGURE 28 COMBINED CYCLE NET SAVINGS VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE
CONDENSER INLET TEMPERATURE = 45°F

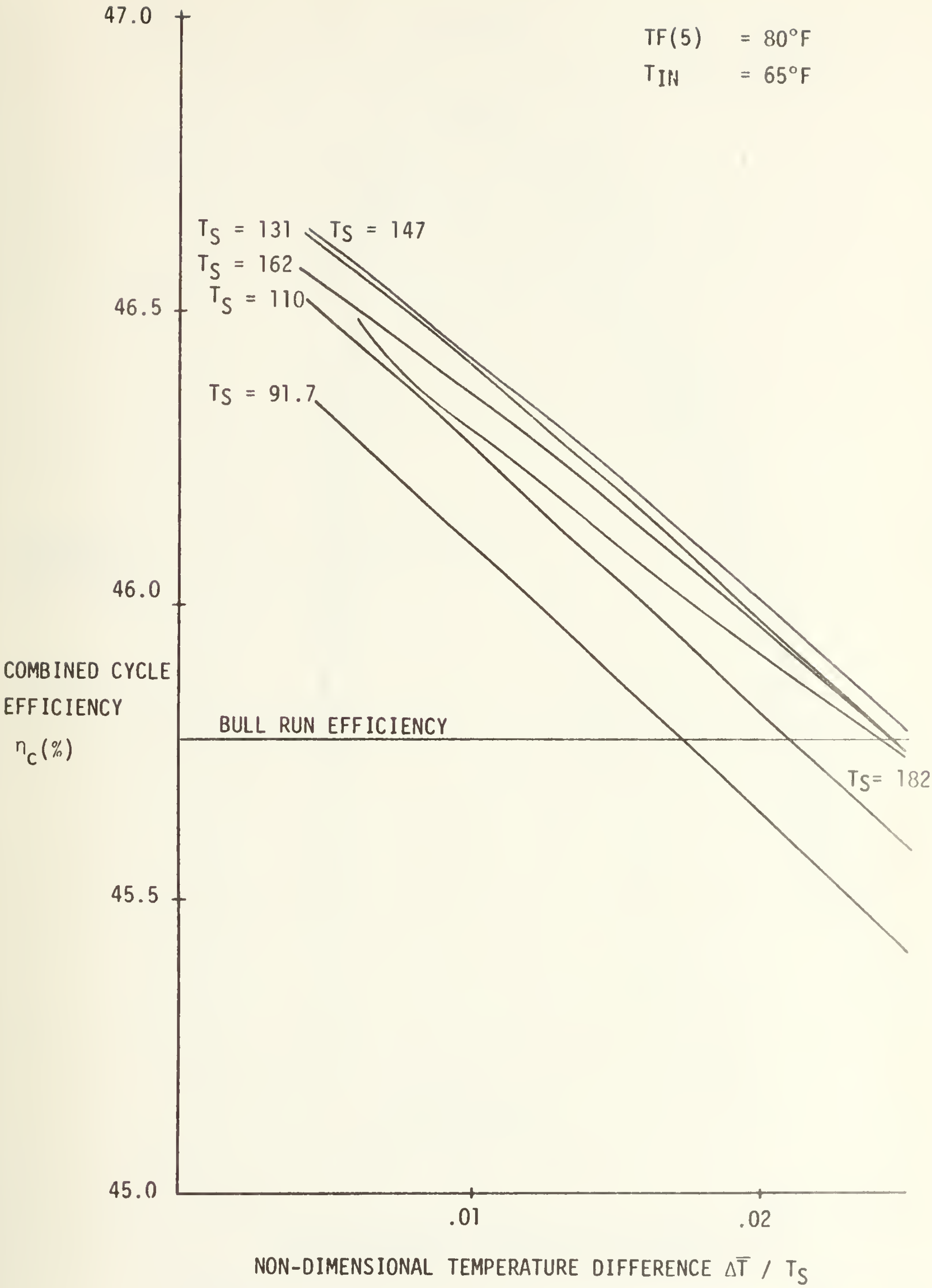


FIGURE 29 COMBINED CYCLE EFFICIENCY VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE
CONDENSER INLET TEMPERATURE = 65°F

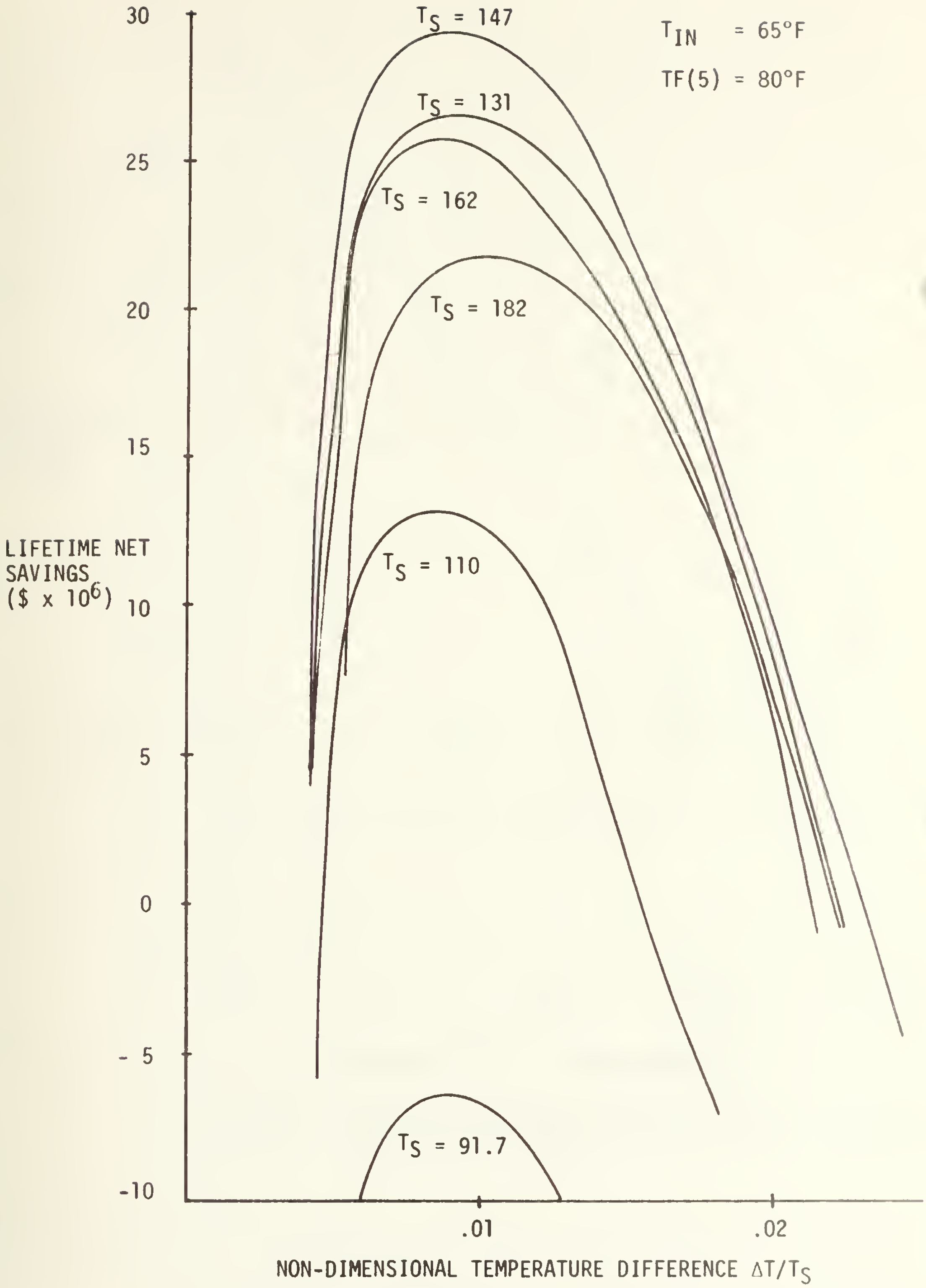


FIGURE 30 COMBINED CYCLE NET SAVINGS VERSUS NON-DIMENSIONAL TEMPERATURE DIFFERENCE
CONDENSER INLET TEMPERATURE 65°F

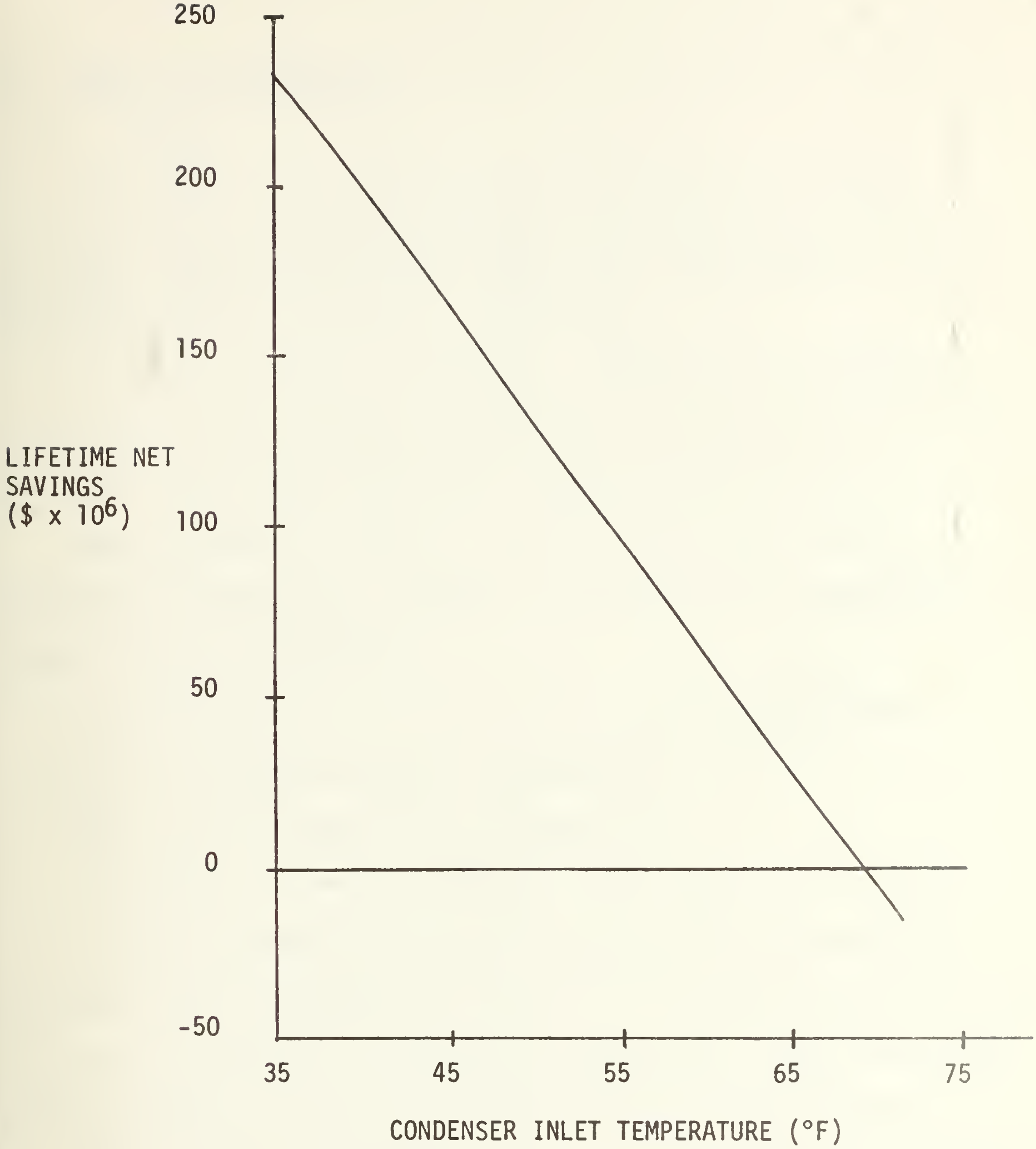


FIGURE 31 OPTIMUM DESIGN NET SAVINGS VERSUS CONDENSER INLET TEMPERATURE

IX. SUMMARY AND CONCLUSION

a. SUMMARY

The purpose of a bottoming cycle is to improve the efficiency of a power plant by enabling the temperature of heat rejection to be lower than what is practical for a steam plant.

By improving the cycle efficiency, a bottoming cycle equipped power plant will consume less fuel, reject less heat to the environment, and deposit fewer harmful products of combustion into the air.

Ammonia seems to be the best choice as the working fluid for a bottoming cycle designed to operate in sub-position to a conventional steam plant such as 'Bull Run'.

The bottoming cycle is defined by three operating parameters:

(1) T_S, the steam condensing temperature.

(2) TF(1), the ammonia boiling temperature.

(3) TF(5), the ammonia condensing temperature.

In order to determine the operating parameters of the optimum bottoming cycle, a thermodynamic analysis was conducted to find the cycle efficiency as a function of the operating parameters. Next, the steam condenser/ammonia boiler and turbomachinery sizes were calculated.

These operating and capital expenses are combined into an equation which yields the net lifetime savings of a combined cycle plant over the unmodified steam plant. The set of operating parameters which gives the maximum economic benefit is taken as the optimum design.

The steam condensing temperature which gives maximum efficiency is that value which minimizes the total power lost due to wetness in both the steam and ammonia cycles.

Limitations on the ammonia condensing temperature are due to corrosion and pumping power considerations on the water side of the condenser. The ammonia condensing temperature for optimum performance is 15°F above the condenser water inlet temperature.

Thus, the economic analysis hinges on the selection of the ammonia boiling temperature, or more precisely the difference between the steam condensing temperature and the entropy averaged temperature of heat addition to the ammonia. As this temperature difference decreases, the thermodynamic efficiency of the combined cycle increases but the area of the steam condenser/ammonia boiler increases. The economic analysis indicates what value of this temperature difference makes the addition of an ammonia bottoming cycle most attractive.

Given the present economic factors enumerated in table 4, the optimum bottoming cycle to be installed at the 'Bull Run' site has the following operating and economic parameters:

(1)	Steam condensing temperature	T_S	= 147°F
(2)	Ammonia boiling temperature	TF(1)	= 142°F
(3)	Ammonia condensing temperature	TF(5)	= 70°F
(4)	Condenser water inlet temperature	T_{IN}	= 55°F
(5)	Cycle net power	P	= 1000 MW
(6)	Overall cycle efficiency	η_{OA}	= 39.1%

(7) Cycle efficiency increase over 'Bull Run' $\Delta\eta$ = 1.4%

(8) Lifetime net savings $98 million or .047¢/kw-hr

(9) Added initial capital cost $9.1 million

b. CONCLUSION

The ammonia bottoming cycle is economically and technologically feasible using existing technology and based on current economic conditions, provided the condenser water inlet temperature is less than 69°F. If the cost of fuel continues to increase, which is likely, the bottoming cycle becomes even more attractive.

The addition of a bottoming cycle seems to be suitable for large conventional utility steam plants, and pressurized-water type nuclear plants employing water-cooled condensers. However, due to the large size of the steam condenser/ammonia boiler which is on the order of 1.5×10^6 square feet of surface area, the bottoming cycle is not deemed practical for applications where power plant size is a major consideration. Additionally, the possibility of a major ammonia leak precludes the installation in a closed-in environment such as a ship.

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APPENDIX I THE COMPUTER PROGRAM

a. INTRODUCTION - The computer program used in the optimization/analysis was run on the M70 and M80 computers at the joint Mechanical/Civil Engineering computer facility at MIT. The programming language used was Fortran IV.

b. IDENTIFICATION OF COMPUTER VARIABLES

VARIABLE	REPRESENTATION
A	Intermediate variable which sums AB's
AB	Area of one quality region of boiling section in SC/AB
ANB	Area of non-boiling section of SC/AB
AT	Total area of steam condenser/NH_3 boiler
A2	NH_3 condenser area
A5	Difference between density of saturated liquid and saturated vapor of NH_3 at condensing temperature
A6	Density of saturated liquid NH_3 at condensing temperature
A7	Difference between NH_3 condensing temperature and the tube wall temperature
BETA	Part of condensing heat transfer coefficient
B1	Average quality in a section of the boiling region of NH_3 boiler
C	Heat transfer coefficient on boiling side of SC/AB (first approximation)
CAPT	Additional capital cost of bottoming cycle
CP	Specific heat at constant pressure of NH_3
C1	Heat transfer coefficient on boiling side of SC/AB(final value)
C3	Heat transfer coefficient on NH_3 side of non-boiling section of SC/AB
C4	Heat transfer coefficient on steam side of non-boiling section of SC/AB (first approximation)
C5	Error between C3 and C4 in iteration loop

VARIABLE	REPRESENTATION
C6	Final value of steam condensing heat transfer coefficient
D	Heat transfer coefficient on condensing side of SC/AB (first approximation)
DELTA	Difference between the steam condensing temperation (T_S) and the average temperature of NH_3 heat addition (TBAR)
DP	Pressure drop on water side of NH_3 condenser
D1	Heat transfer coefficient on condensing side of SC/AB (final value)
D2	NH_3 boiler tube diameter
D3	NH_3 condenser tube diameter
E	Error in heat flux calculation in SC/AB (C-D)
EFF	Thermal efficiency of NH_3 cycle
EFFOA	Thermal efficiency of combined cycle excluding condensing pump requirements
EFFST	Efficiency of steam cycle at specified steam condensing temperature
ENTH	Corrected enthalpy used in NH_3 condensing calculation
ETA	Overall thermal efficiency of combined cycle
E1	Absolute difference between first and second approximation in NH_3 condenser water outlet temperature routine
E2	Error in first approximation of heat flux in NH_3 condenser
E3	Absolute value of E2
F	Chen factor for boiling
FI( )	Intermediate function used to calculate HG( )
FII( )	Intermediate function used to calculate SG( )
G	NH_3 mass flow per unit area in NH_3 boiler
GW	Mass flow rate/unit area of water in NH_3 condenser
H( )	Enthalpy (NH_3)
HC	Convective component of the heat transfer coefficient in the NH_3 boiler
HCOND	Condensing NH_3 heat transfer coefficient
HF( )	Enthalpy of saturated liquid (NH_3)
HFG( )	Latent heat of evaporation (NH_3)

VARIABLE	REPRESENTATION
HG( )	Latent heat of evaporation vapor (NH_3)
HNUB	Nucleate boiling component of heat transfer coefficient in NH_3 boiler
HS( )	Enthalpy (based on isentropic expansion in NH_3 turbine)
HW	Convective heat transfer coefficient of water side of NH_3 condenser
I	Counting integer in DO LOOP
IR	Capitalization rate
KF	Thermal conductivity of NH_3 in NH_3 boiler
KF1	Thermal conductivity of NH_3 in NH_3 condenser
KW	Thermal conductivity of water
L	Temperature step side in ΔT variation routine
LMTD	Log mean temperature difference SC/AB
ME( )	NH_3 turbine mass flow extraction for feedheating
MNH_3	Mass flow rate through NH_3 boiler
MW	Mass flow rate through NH_3 condenser
N	Counting integer in DO LOOP
NM	Turbine efficiency after mechanical, exit and stage losses
NOT	Number of tubes in NH_3 boiler
NP	Pump efficiency
NS	Isentropic efficiency of turbines
P( )	Pressure (in thermodynamic properties of NH_3 sub routine)
PF	Density of NH_3 saturated liquid
PG	Density of NH_3 saturated vapor
PH2O	Power produced by steam cycle
PNH3	Power produced by NH_3
PP	NH_3 condenser water pumping power
PUMPW	Total NH_3 feed pumping power
PX( )	Intermediate function used to calculate P( )
Q	Heat added to NH_3 cycle
QA	Enthalpy added to NH_3 in SC/AB

VARIABLE	REPRESENTATION
QAD	Heat added to steam cycle
QB	Total heat added to boiling section of NH_3 boiler
QNB	Total heat added to non-boiling section of NH_3 boiler
QR	Total heat rejected by combined cycle
RETP	Two-phase flow Reynolds number
R1	Heat exchanger cost $/FT^2
R2	Steam turbine cost $/MW
R3	NH_3 turbine cost $/MW
R4	Boiler efficiency
R5	Bull Run thermal efficiency
R6	Coal cost $/TON
R7	Generator efficiency
R8	Load factor
R9	Plant life and loan duration
S( )	Entropy
SAVE	Lifetime savings gained by bottoming cycle equipped plant over base case
SF( )	Entropy of saturated liquid (NH_3)
SG( )	Entropy of saturated vapor (NH_3)
ST	Surface tension of NH_3 in NH_3 boiler
S1	Boiling suppression factor in two-phase flow
T( )	Temperature (°R)
TAV	Average water temperature in NH_3 condenser
TBAR	Average temperature of heat addition of NH_3
TF( )	Temperature (°F)
TIN	Inlet water temperature in NH_3 condenser
TOUT	Outlet water temperature in NH_3 condenser (first approximation)
TOUTB	Outlet water temperature in NH_3 condenser (second approximation)
TS	Steam condensing temperature (°F)
TURBW	Total NH_3 turbine power

VARIABLE	REPRESENTATION
TT	NH_3 condenser tube wall temperature
TW	SC/AB tube wall temperature
U	Overall heat transfer coefficient
UB	Overall average heat transfer coefficient in boiling section of the SC/AB
UC	Overall heat transfer coefficient in NH_3 condenser
UF	Absolute viscosity of NH_3 saturated liquid in NH_3 boiler
UF1	Absolute viscosity of NH_3 saturated liquid in NH_3 condenser
UG	Absolute viscosity of NH_3 saturated vapor in NH_3 boiler
UNB	Overall heat transfer coefficient in non-boiling section of the SC/AB
UW	Average absolute viscosity of water in NH_3 condenser
VF( )	Specific volume of NH_3 saturated liquid
VFG( )	VG( ) - VF( )
VG( )	Specific volume of NH_3 saturated vapor
VW	Velocity of water in NH_3 condenser
WP( )	NH_3 feed stage pump work
WT( )	NH_3 turbine stage work
X( )	Quality
XS( )	Quality based on isentropic expansion in NH_3 turbine
XTT	Martinelli parameter for boiling
YTT	Inverse of XTT
Y	% quality in NH_3 boiler section (B1 x 100)
Z	Delta/T_S(°R)

c. PROGRAM DESCRIPTION

For clarity, the computer program on the following pages is divided into nine zones, labelled 'A' through 'I'. Each zone represents a distinct function in the optimization routine.

Zone 'A' includes the identification of variables as either integers or real numbers. The 'dimension' statements reserve core space for the subscripted variables.

Zone 'B' consists of the engineering and economic factors listed in table four. It is these factors which the economic analysis hinges upon. Accordingly, these parameters are easily changed to allow an up-to-date analysis. Also included in zone 'B' are the ammonia condensing temperature and the coolant water inlet temperature for the plant under consideration.

The steam condensing temperature, steam cycle efficiency, and part of the steam side heat transfer coefficient are included in zone 'C'. Also included are the temperature difference between the condensing steam and boiling ammonia, as well as the pump and turbine efficiencies.

Part 'D' establishes the three intermediate temperature levels in the ammonia cycle and generates the thermodynamic properties of ammonia needed for the analysis. The equation of state for the ammonia are empirical curves, fitted to the tabulated data[24].

The thermodynamic analysis of the ammonia cycle and the calculation of the combined cycle thermal efficiency occurs in zone 'E'. It is in this section where the non-dimensional temperature difference between the condensing steam and boiling ammonia is calculated.

In zone 'F' the heat transfer properties of the ammonia are calculated from empirical relations.[25]

Zone 'G' is the most complex part of the entire program. In this section the steam condenser/ammonia boiler area is calculated. Note that in statements 55 and 77 the tube wall temperature is incrementally changed. As stated in chapter four, the heat flux equation for both sides of the steam condenser/ammonia boiler can only be solved by interating until the wall temperature which satisfies the equality is found. In order to minimize the running time of the program, an exact equality of both sides of equation (34) is not demanded, but rather it is required that the two results are within 1% of each other. Without this relaxation of the solution, the temperature increment required to obtain a converging iterative solution would increase the running time and cost of the program by several orders of magnitude. Note also that the heat exchanger analysis is repeated for the 10 quality regions discussed in chapter four, as well as for the non-boiling region.

The ammonia heat transfer properties and condenser pumping power are computed in zone 'H'. Once again the tube wall temperature is found by iteration.

In zone 'I', the condenser pumping power is incorporated into the cycle efficiency. The initial capital, operating, and lifetime costs are also calculated in this region. The output statements comprise the final section of the program.

```
      REAL
      REAL IFIRST,A2,TS,EFFCA,TEAR,LMTD,A,A1,U,DELTA,.
      REAL NS,NF,NM,NP
      REAL A5,A6,A7
      REAL TURBW,PUMPW,QF
      REAL CAPT
      REAL I
      REAL NOT                                                 SECTION A
      REAL IR,R1,R2,R3,R4,R5,R6,R7,R8,R9,SAVE
      REAL DF,CW,HW,KW,TAV,E2,HCOND
      REAL E1,TOUT,TOUTB,GW,MW,D3,QF,QAD,TIN,UF1,KF1,UC,ETA,PP
      REAL RE,RA,CAPI
      REAL VA,ENTH,PNH3,PH20
      REAL S1,MNH3,G,ST,CF,KF,UG,HF,PG,PF,PNUP,HC,RETP,C,D,E,BETA
     1UB,GR,CNT,C1,D1,AB,ANB,AT,UNB,C3,C4,C5,C6,B,B1,XTT,YTT,D2,TW,F
      INTEGER M
      INTEGER N
      INTEGER I
      DIMENSION S(50),H(50),X(50),XS(50),ME(50),WP(50)
      DIMENSION TF(50),T(50),PX(50),P(50),VF(50),VG(50),SF(50),SG(50),
     1HF(50),HFG(50),HG(50),DPDT(50),FI(50),FII(50)
      DIMENSION VFG(50)
      DIMENSION WT(50)
      DIMENSION HS(50)
```

```
C     INTEREST RATE
      IR=.12
C     HEAT EXCHANGER COST PER SQ FT
      R1=3.
C     STEAM TURBINE COST $/MW
      R2=31000.
C     NH3 TURBINE COST $/MW
      R3=5000.
C     BOILER EFF                                               SECTION B
      R4=.89
C     UNMODIFIED CYCLE EFF
      R5=.4578
C     COAL COST $/TON
      R6=50.
C     GENERATION EFF
      R7=.99
C     LOAD FACTOR
      R8=.8
C     PLANT LIFE & LOAN DURATION
      R9=30.
C     AUX EFFICIENCY
      RA=.94
C     HIGHER HEATING VALUE OF FUEL
      PF=10788.
C     NUMBER OF TUBES PER BANK NH3 BOILER
      NOT=70.
```

SECTION B

```
C       TUBE DIAMETER NH3 BOILER
        D2=1./12.
C       NH3 CONDENSING TEMP
        TF(5)=70.
C       INLET COOLING WATER TEMP
        TIN=TF(5)-15.
C       TUBE DIAMETER NH3 CONDENSOR
        D3=.875/12.
C       NH3 CONDENSOR AREA
        A2=3.556E 05
```

SECTION C

```
C       FUNCTIONS OF STEAM CONDENSING TEMP
        TS=147.
        BETA=3917.2/(NOT**.25)
        LFEST=.4194
        L=2.5
   22   IF (L.GT.20.) GO TO 310
        TF(1)=TS-L
        L=L+2.5
        I=3
K       MECHANICAL,EXIT,& STAGE LOSSES
        NM=(.99)**3
C       ISENTROPIC EFF
        NS=.84
C       PUMP EFF
        NP=.9
```

SECTION D

```
        TF(2)=TF(5)+.75*(TF(1)-TF(5))
        TF(3)=TF(5)+(TF(2)-TF(5))*(2./3.)
        TF(4)=(TF(5)+TF(3))/2.
C       THERMODYNAMIC PROPERTIES OF NH3
        N=1
        DO 99 N=1,5,1
        T(N)=TF(N)+459.7
        PX(N)=25.5743247-(3295.1254/T(N))-(6.401247*ALOG10(T(N)))
       1-(4.148279E-04)*T(N)+(1.4759945E-06)*(T(N))**2
        P(N)=10.**PX(N)
        FI(N)=(5300.-32.*P(N)+.10132*P(N)**2-9.92E-05*P(N)**3)*(1.E-08)
        VG(N)=FI(N)*T(N)+.6301952*(T(N)/P(N))-((3.18228E 07/T(N)**3)+((
       23.80226E 27+2.29909E 26*P(N))/T(N)**11)+((1.1778E 38*P(N)**5)/
       3T(N)**19))-.041648
        IF (TF(N).LE.175.) GO TO 13
        HG(N)=76.0959*ALOG10(T(N))-P(N)*(2.3577E 07/T(N)**3+(8.451E 27+
       12.555E 26*P(N))/T(N)**11+(7.272E 37*P(N)**5)/T(N)**19)-.007714*
       2P(N)+.269065*T(N)+(1.58047E-04)*T(N)**2+262.303
       3+(TF(N)-175.)**2/25.
        GO TO 15
   13   HG(N)=76.0959*ALOG10(T(N))-P(N)*(2.3577E 07/T(N)**3+(8.451E 27+
       12.555E 26*P(N))/T(N)**11+(7.272E 37*P(N)**5)/T(N)**19)-.007714*
       2P(N)+.269065*T(N)+(1.58047E-04)*T(N)**2+262.303
   15   FII(N)=P(N)*(982.-2.964*P(N)+.006255*P(N)**2-4.59E-06*P(N)**3)*
       1(-1.E-08)
        SG(N)=.619546*ALOG10(T(N))-(P(N))/T(N)*((1.7683E 07/T(N)**3)+
       1(7.747E 27+2.3421E 26*P(N))/T(N)**11+(6.908E 37*P(N)**5)/
       2T(N)**19)-.2687723*ALOG10(P(N))+(3.16094E-04*T(N))-33.048/
```

```
4I(N)+.028463+FII(N)
 VF(N)=(.0680064+3.7073E-03*SQRT(271.4-TF(N))-7.3732E-05*
1(271.4-TF(N)))/(1.+.31663*SQRT(271.4-TF(N))+8.8544E-03*
2(271.4-TF(N)))
 VFG(N)=VG(N)-VF(N)
 HF(N)=75.7+1.135*(TF(N)-30.)                                SECTION D
3+(ABS(TF(N)-170.)+(TF(N)-170.))*.25
 HFG(N)=HG(N)-HF(N)
 SF(N)=SG(N)-(HFG(N)/T(N))
99 CONTINUE
```

```
 N=2
C     THERMODYNAMIC ANALYSIS
 DO 111 N=2,5,1
 WP(N)=VF(N)*(P(1)-P(5))/4.*(144./778.)*(1./NP)
 S(1)=SG(1)
 H(1)=HG(1)
 X(1)=1.
 XS(N)=(S(N-1)-SF(N))/(SG(N)-SF(N))
 HS(N)=HF(N)+XS(N)*HFG(N)
 H(N)=H(N-1)-(H(N-1)-HS(N))*NS
 X(N)=(H(N)-HF(N))/HFG(N)
 S(N)=SF(N)+X(N)*(SG(N)-SF(N))                               SECTION E
111 CONTINUE
 ME(1)=(HF(2)-HF(3)-WP(3)) /(H(2)-HF(3)-WP(3))
 ME(2)=(HF(3)-(1.-ME(1))*(HF(4)+WP(4)))/(H(3)-HF(4)-WP(4))
 ME(3)=(HF(4)-(1.-ME(1)-ME(2))*(HF(5)+WP(5)))/(H(4)-HF(5)-WP(5))
 QA=HG(1)-HF(2)-WP(2)
 PUMPW=WP(2)+(1.-ME(1))*WP(3)+(1.-ME(1)-ME(2))*WP(4)+
1(1.-ME(1)-ME(2)-ME(3))*WP(5)
 N=1
 DO 200 N=1,4,1
 WT(N)=(H(N)-H(N+1))*NM
200 CONTINUE
 TURBW=WT(1)+(1.-ME(1))*WT(2)+(1.-ME(1)-ME(2))*WT(3)+
1(1.-ME(1)-ME(2)-ME(3))*WT(4)
 EFF=(TURBW-PUMPW)/QA
 TBAR=(TF(1)*HFG(1)+((TF(1)+TF(2))/2.)*(HF(1)-(HF(2)+WP(2))))/
1(HG(1)-HF(2)-WP(2))
 DELTA=TS-TBAR
 EFFOA=EFFST+(1.-EFFST)*EFF
 LMTD=(.25*(TF(1)-TF(5)))/ALOG((TS-.75*TF(1)-.25*TF(5)) /
1(TS-TF(1)))
 Z=DELTA/(TS+459.7)
```

```
C     HEAT TRANSFER PROPERTIES
 QAD=(1000.*3.413E 06)/EFFOA
 Q=QAD*(1.-EFFST)
 MNH3=Q/(H(1)-HF(2)-WP(2))                                   SECTION F
 G=(4.*MNH3)/(DE**2*3.14159*5000.)
 UF1=.286-.001395*(TF(5)-107.)
 KF1=.263-.000748*(TF(5)-97.)
 ST=12.11-.0793*(TF(1)-97.)
 CL=1.155+.00193*(TF(1)-97.)
 UG=.0287+6.484E-05*(TF(1)-97.)
```

```
      UF=.286-.001395*(TF(1)-107.)
      PG=1./VG(1)
      PF=1./VF(1)                                              SECTION F
      KF=.263-.000746*(TF(1)-97.)
C     NHB BOILER COMPUTATIONS
      BNHB=(25.6868*(KF**.79)*(CF**.45)*(PF**.49)*((HFG(1))**.51))/
     1((ST**.5)*(UF**.29)*(PG**.24)*((VFG(1))**.75)*((T(1))**.75))
      HC=.023*(KF/D2)*((G*D2/UF)**.8)*((CP*UF/KF)**.4)
      TW=(TF(1)+TF(2))/2.
   55 TW=TW+.01
      IF (TW.GT.TS) GO TO 307
      C3=HC*(TW-(TF(1)+TF(2))/2.)
      C4=BETA*(TS-TW)**.75
      C5=ABS(C3-C4)
      IF (C5.GT.25.) GO TO 55
      C6=C4/(TS-TW)
      UNB=(HC*C6)/(HC+C6)
      QNB=Q*(HF(1)-HF(2)-WF(2))/(H(1)-HF(2)-WF(2))
      ANB=QNB/(UNB*(TS-(TF(1)+TF(2))/2.))
      A=0.
      I=5
      DO 88 I=5,95,10                                          SECTION G
      Y=FLOAT(I)
      B1=Y/100.
      B=Y/5.
      XTT=(((1.-B1)/B1)**.9)*((PG/PF)**.5)*((UF/UG)**.1)
      YTT=1./XTT
      IF (YTT.GT.1.) GO TO 66
      F=10.**((ALOG10(YTT)-1.)/3.419)
      GO TO 67
   66 F=10.**(((ALOG10(YTT)-ALOG10(.4))/1.4367)+ALOG10(1.5))
   67 RETP=(G*(1.-B1)/UF)*F**1.25
      S1=.84-((ALOG10(RETP)-4.301)/1.8324)
      TW=TF(1)
   77 TW=TW+.01
      IF (TW.GT.TS) GO TO 307
      C=(BNUB*(TW-TF(1))**.99*S1+HC*(1.-B1)**.8*F)*(TW-TF(1))
      D=BETA*(TS-TW)**.75
      E=ABS(C-D)
      IF (E.GT.25.) GO TO 77
      C1=C/(TW-TF(1))
      D1=D/(TS-TW)
      UB=(C1*D1)/(D1+C1)
      QB=Q-QNB
      AB=(.1*QB)/(UB*(TS-TF(1)))
      A=AB+A
   88 CONTINUE
      UB=QB/((TS-TF(1))*A)
      AT=A+ANB
      WRITE (5,1)
    1 FORMAT ('  TS',6X,'TF1',5X,'EFFOA',5X,'Z',7X,'UNB',7X,'UB',
     19X,'ANB',9X,'A',9X,'AT')
      WRITE (5,275) TS,TF(1),EFFOA,Z,UNB,UB,ANB,A,AT
```

```
275 FORMAT ( F6.1,2X,F6.1,2X,F7.5,2X,F7.5,2X,F7.2,2X,F7.2,2X,
   1F10.2,2X,F10.2,2X,F10.2)                                    SECTION G
C     NH3 CONDENSOR DESIGN
      TOUT=TIN+5.
   44 TOUT=TOUT+.01
      TAV=(TOUT+TIN)/2.
      UW=4.33-.0579*(TAV-32.)
      KW=.319+.00066*(TAV-32.)
      QF=QAD*(1.-EFFSA)
      MW=QF/(TOUT-TIN)
      GW=(4.*MW)/(D3**2*3.14159*41368.)
      VW=GW/224280.
      HW=.023*(KW/D3)*(GW*D3/UW)**.8*(UW/KW)**.4
      TT=TAV
   40 TT=TT+.005
      IF (TT.GT.TF(5)) GO TO 307                                SECTION H
      IF (TT.LT.TAV) GO TO 307
      ENTH=HFG(5)+.426*(TF(5)-TT)
      A5=1./VF(5)-1./VG(5)
      A6=1./VF(5)
      A7=TF(5)-TT
      HCOND=(1.+(46.31/HFG(5))*(TF(5)-TT))*72.9*
     1(((1./VF(5))*(1./VF(5)-1./VG(5))*(KF1**3)*ENTH)/(D3*UF1*
     2(TF(5)-TT)))**.25
      E2=HW*(TT-TAV)-HCOND*(TF(5)-TT)
      E3=ABS(E2)
      IF (E3.GT.20.) GO TO 40
      UC=(HW*HCOND)/(HW+HCOND)
      TOUTE=2.*(TF(5)-(QF/(UC*A2)))-TIN
      E1=ABS(TOUT-TOUTE)
      TOUT=TOUT+(TOUTE-TOUT)/10.
      IF (E1.GT..5) GO TO 44
      DP=(1.9679E-11)*(GW**2)*((GW*D3)/UW)**-.25
      PP=(DP*MW/62.3)*(144./778.)*(1./3.413E 06)
C     OVERALL THERMAL EFFICIENCY
      ETA=(3.413E 06*(1000.-PP))/QAL
C     POWER SPLIT
      PH20=(QAD*EFFST)/(3.413E 06)                              SECTION I
      PNH3=1000.-PH20
      WRITE (5,444)
  444 FORMAT ('0','  PF',7X,'TIN',5X,'TOUT',4X,'EFF*',6X,'VW',4X,
     1'PH20',5X,'PNH3',3X,'HHV')
      WRITE (5,450) PP,TIN,TOUT,ETA,VW,PH20,PNH3,RF
  450 FORMAT ( F7.2,2X,F6.1,2X,F6.1,2X,F7.5,3X,F6.1,
     12X,F6.1,2X,F6.1,2X,F7.1)
C     COST EQUATION
      CAPT=R1*AT-PNH3*(R2-R3)
      CAPI=CAPT*R9*(IR*((1.+IR)**R9)/((1.+IR)**R9-1.))
      SAVE=((1.4949E 10*R6*R8*R9)/(RA*RF*R4*R7))*(1./R5-1./ETA)-CAPI
      WRITE (5,451)
  451 FORMAT ('0','  INTEREST',2X,'HEATEX $/FT2',2X,'$/MW H20',2X,
     1'$/MW NH3',2X,'BOILER EFF',2X,'AUX EFF')
      WRITE (5,452) IR,R1,R2,R3,R4,RA
```

```
 203C8      452 FORMAT ( 3X,F5.3,8X,F5.3,6X,F7.1,2X,F7.1,6X,F5.3,6X,F5.3)
 2076A          WRITE (5,453)
 208A8      453 FORMAT ('0','  UNMOD EFF',2X,'CL COST',2X,'GEN EFF ',2X,
 208A8         1'LOAD FACTOR',6X,'CAP COST',7X,'SAVINGS')
 20EA4      454 FORMAT ( 3X,F7.5,3X,F7.2,2X,F7.3,7X,F5.3,2X,F15.2,2X,F15.2)
 2124E          WRITE (5,454) R5,R6,R7,R8,CAPT,SAVE
 2168B          WRITE (5,455)
 217CB      455 FORMAT ('  ********************************************')
 2182E      304 GO TO 22
 2186C      305 GO TO 310
 21AAF      307 WRITE (5,308)
 21CBA      308 FORMAT ('  NON-CONVERGING SOLUTION')
 21F2A      310 END
0000[S] .U      21F6[V] Z       21FA[V] EFFST   21FE[V] A2      2202[V] TS
2206[V] EFFOA   220A[V] TBAR    220E[V] LMTD    2212[V] A       2216[V] A1
221A[V] U       221E[V] DELTA   2222[V] Q       2226[V] NS      222A[V] NP
222E[V] NN      2232[V] NB      22FA[V] A5      22FE[V] A6      2302[V] A7
2306[V] TUFPW   230A[V] IUFPW   230E[V] QA      2312[V] CAPT    2316[V] L
231A[V] NOT     231E[V] IR      2322[V] R1      2326[V] R2      232A[V] R3
232E[V] R4      2332[V] R5      2336[V] R6      233A[V] R7      233E[V] R8
2342[V] R9      2346[V] SAVE    234A[V] DP      234E[V] UW      2352[V] HW
2356[V] KW      235A[V] TAV     235E[V] E2      2362[V] HCOND   2366[V] E1
236A[V] TOUT    236E[V] TOUTF   2372[V] GW      2376[V] MW      237A[V] D3
237E[V] QR      2382[V] QAD     2386[V] TIN     238A[V] UF1     238E[V] KF1
2392[V] UC      2396[V] ETA     239A[V] PP      239E[V] FF      23A2[V] RA
23A6[V] CAPI    23AA[V] VW      23AE[V] ENTP    23B2[V] PNH3    23B6[V] PH2O
23BA[V] S1      23BE[V] MNH3    23C2[V] G       23C6[V] ST      23CA[V] CP
23CE[V] KF      23D2[V] UG      23D6[V] UF      23DA[V] PG      23DE[V] PF
23E2[V] ENUT    23E6[V] HC      23EA[V] BETF    23EE[V] C       23F2[V] D
23F6[V] E       23FA[V] BETAUB  23FE[V] QR      2402[V] QNB     2406[V] C1
240A[V] D1      240E[V] AB      2412[V] ANP     2416[V] AT      241A[V] UNB
241E[V] C3      2422[V] C4      2426[V] C5      242A[V] C6      242E[V] B
2432[V] E1      2436[V] XTT     243A[V] YTT     243E[V] D2      2442[V] TW
2446[V] F       244A[V] X       244E[V] N       2452[V] I       2456[V] S
251E[V] H       2516[V] X       26AE[V] XS      2776[V] WP      283E[V] TF
2906[V] T       29CE[V] PX      2A96[V] P       2B5E[V] VF      2C26[V] VG
2CEE[V] SF      2DB6[V] SG      2E7E[V] HF      2F45[V] HFG     300F[V] HG
30D6[V] DPLT    319E[V] FI      3266[V] FII     332F[V] VFG     33F6[V] WT
34BE[V] HS      35DA[V] PETA    0000[S] .A      00D8[L] 22      21F2[L] 310
0000[S] .R      0C58[L] 99      0000[S] SZ      0000[S] ALOG10  0660[L] 13
07E8[L] 15      0000[S] SQRT    0000[S] ABS     0FFF[L] 111     10F4[L] 200
3736[V] EFE     0000[S] ALOG    1504[L] 55      21FA[L] 307     185E[L] 88
37A6[V] Y       0000[S] FLOAT   16EC[L] 66      172E[L] 67      177E[L] 77
37CE[V] UB      190A[L] 1       0000[S] @I      19C6[L] 275     1A22[L] 44
37EA[V] TT      1B12[L] 40      37FE[V] E3      1DD2[L] 444     1E7A[L] 450
1F95[L] 451     203E[L] 452     208A[L] 453     20EA[L] 454     217C[L] 455
2182[L] 304     2186[L] 305     21CF[L] 308     0000[S] .V
  THIS PROGRAM WILL OCCUPY X'3B12' BYTES OF STORAGE SPACE
  PROGRAM  *MAIN* HAS    NO ERRORS
// XEQ
 EOF
```

APPENDIX II STEAM CYCLE DATA

The computer program discussed in Appendix I requires the following information about the steam cycle as inputs:

(1) The steam condensing temperature, T_S.

(2) The steam cycle thermal efficiency at T_S.

(3) The portion of the steam side heat transfer (condensing) coefficient in the steam condenser/ammonia boiler, which is a function of T_S i.e. the fluid properties.

Carrying out the modification of the 'Bull Run' plant detailed in chapter five, the steam cycle thermal efficiency versus the steam condensing temperature is calculated.

The properties involved in the calculation of the steam side heat transfer coefficient and their relationship are shown in equation (23). The portion of equation (23) which includes only the fluid properties is called 'β_0'.

Table A2-1 shows values of thermal efficiency, ηth_{steam} and 'β_0' for the different values of steam condensing temperature used in the analysis.

T_S(°F)	ηth_{steam}	β_0(BTU/(hr-ft-(°F)$^{.75}$))
91.7	.4578	3438.6
110	.4451	3642.4
127	.4336	3760.3
131	.4308	3791.7
147	.4194	3917.2
162.2	.4084	4085.5
182.2	.3952	4270.1

TABLE A2-1 VARIATION OF STEAM CYCLE EFFICIENCY AND 'β_0' WITH THE STEAM CONDENSING TEMPERATURE